TASK CORPORATION II (A)

Failure of a Ball Bearing

In September, 1963, Mr. Jack Wireman, an engineer at Task Corporation was faced with the question of what should be done about the failure of an electric motor shaft bearing. The motor was a new design to power an aircraft hydraulic pump. A contract for 300 such motors had been placed with Task, and the terms of this contract required that the motor be capable of operating for 2,500 hours without failure. When the motor in test began to draw excessive current after only 1,800 hours operation, it was shut down and disassembled. Examination showed severe pitting and wear of the balls and races in the front shaft bearing. Pictures of the bearing after failure appear in Exhibit 1. Cross section drawings of the motor and the hydraulic pump to which it attached appear in Exhibits 2 and 3.

Task was at this time a company of 140 employees in Anaheim, California which manufactured a variety of products. These included electric motors, pumps, fans, blowers, refrigeration systems, and measuring instruments for wind tunnel testing. The electric motors ranged in size from fractions to hundreds of horsepower and were for special applications where high performance such as great speed, low weight or small volume per horsepower were required. Confidence in Tasks' ability to design high performance motors was reflected in a comment by Mr. Elmer Ward, the company's president: "I once told a man that he could have anyone he wanted design him an electric motor, and if we couldn't come up with a better one in 24 hours, I'd pay one hundred dollars. He never collected, and the offer is still open."

In April, 1962, Task was invited by the Geyser Pump Company to bid on the job of designing and producing an electric motor to be used for driving an aircraft hydraulic pump. Geyser was, in turn, preparing its own bid to make the pump, attach the motor to the pump, and sell the completed units to the Thunder Aircraft Company for installation in Thunder 99 aircraft. Two such units were required per aircraft. Each pump would operate continuously during flight. A specification described the pump as "the primary source for the flight power control units as well as an alternate source of hydraulic pressure. It is also used as a hydraulic pressure source for use during pre-flight testing and routine aircraft maintenance operations." In addition to these electrically powered hydraulic pumps, each 99 had its main pumps which were driven mechanically by the turbojet engines.

Prepared in the Design Division, Department of Mechanical Engineering, Stanford University, by Karl H. Vesper as a basis for student engineering problems. Assistance from Jack Wireman, Elmer F. Ward, Dino Morelli and Thomas Barish is gratefully acknowledged.

Operating requirements of the electric pump units included the following:

Hydraulic Fluid	- Skydrol 500A (See fluid specification, Exhibit 4.)
Fluid Temperature	- 65°F to 180°F $\pm$ 5
Inlet Pressure	- 45-50 psi
Outlet Pressure	- 3,000 psi $\pm$ 50
Flow Rate	- 1 to 6 gallons/min. $\pm$ 0.1
Input Power	- 400 cycle A.C. at 200 volts
Output Power (motor only)	- 11.75 horsepower
Weight (motor only)	- 18.75 lbs.
Efficiency (motor only)	- 65%
Maximum Motor Case Temp.	- 225°F
Explosion Proof	
Life	- 2,500 hours with "no failure of parts or excessive wear"

In addition to these operating conditions, the motor was required to comply with eight U.S. Government specification documents, eleven non-government specification documents, and various other requirements relating to torque at different speeds, rotational acceleration at 65°F, flammability, corrosion and fungus resistance, electrical bonding, radio interference, noise level, fume propagation, safety wiring, dielectric arc-over resistance, and fluid leakage under pressure.

Testing of the performance and life was to be done by Geyser Pump on two "qualification units." A schematic diagram of the test rig is shown in Exhibit 5. In the life test, each motor was to be run through several different cycles as described in the schedule of Exhibit 6.

After some discussion between engineers of Task and Geyser it was decided that a 6,000 RPM synchronous unit would allow the best compromises of weight and size. Task engineers planned on the motor being liquid cooled by the hydraulic fluid enroute to the pump. The fluid would, after passing through a ten micron filter, enter one side of the motor at the front, flow back through annular slots around the stator, circulate forward through the rotor, then pass through a centrifugal impeller at the front of the motor and into the Geyser pump for boost up to 3,000 psi. Both front and rear bearings of the motor would be kept fully immersed by the hydraulic fluid as it circulated through to the pump under a pressure within the motor of around 50 psi. A schematic of this flow pattern appears in Exhibit 7. Task offered to design and build the motors at a price between $600. and $1,000. each, depending on the quantity ordered.

Geyser Pump Company won a contract for the motor-pump units from Thunder and gave Task a contract to provide 300 motors in total over a 12 month period. The first two units would be for qualification tests to assure that the motors performed as required. All remaining units would be for installation in Thunder aircraft. It was expected that contracts for more of the motors would be placed with Task in the future since Task was the only company selected to make them. If, however, the motors did not meet performance specifications of the contract, Geyser

could refuse to accept them from Task and would not have to pay for them. Similarly, by the contract between Thunder and Geyser, Thunder could refuse to accept the complete pump units if either the motor or the pump did not measure up to specified performance.

When the Task proposal was accepted, engineers at Task proceeded with detailed design of the motor. Electrical components, numbers of turns, amount of iron and other aspects of the electromagnetic circuitry were determined. Dimensions of the path for circulating the hydraulic fluid and the shape of the centrifugal impeller were designed. The shape and dimensions of the aluminum motor housing had to be arranged to fit the electrical and hydraulic requirements and also to mate with the Geyser pump. Power transmission between motor and pump was to be through a spline, the dimensions of which were given by Geyser. Shaft dimensions of the motor were picked by Task engineers to transmit expected loads with abundant safety and to blend with the required spline.

Three types of loading were expected on the motor shaft, torsional, radial, and axial. Torsional loading would be primarily due to the torque output of the motor. Radial loading would be mainly caused by the weight of the rotor and impeller (5.3 lbs.), and by rotor dynamic imbalance, which by careful manufacture, testing and balancing would be limited to five pounds. The main axial load on the motor shaft was expected to be imposed by the pump shaft which was axially pre-loaded by a spring in the pump. The pump had a male spline which was pushed into the mating motor shaft, compressing the spring, according to Geyser engineers, to a force of 50 to 70 pounds.

Ball bearings made by the Barden Corporation were picked for the shaft. The use of Barden bearings had become something of a standard practice at Task after problems had been encountered with bearings of several other manufacturers. Barden specialized in bearings of high precision. Although relatively expensive, such bearings were considered highly reliable.

The front bearing chosen for the motor was a Barden No. 204SST5 conrad, with an outer race centered[1] cage and a contact angle[2] of 14°. This bearing was chosen because it apparently provided a sufficient margin of

1. "Outer race centered" refers to the manner in which the ball cage or retainer of the bearing is centered. In most commonly used bearings, the ball cage is centered by contact against the inner race. At high rotational speeds, centrifugal force imparted by the cage can make if difficult for lubricant to reach the inner race. Also at high speeds the cage may become centrifugally expanded and thereby lose centering contact. By centering through contact with the outer race, such loss of centering contact is better controlled for many applications.

2. "Contact angle" refers to the angle between a line perpendicular to the shaft axis of rotation and a line through both the center of a ball and the point at which that ball contacts with the outer race. The initial contact angle is defined by assuming contact with no deflections and is fixed by radial looseness.

load capacity and its inner race was large enough to slip easily over the internally splined shaft, and its outer race would fit conveniently into the motor housing. Radial allowances between 0.0001 and 0.0003 inches were provided between inner race and shaft and between outer race and housing. To absorb axial loading on the shaft, the inner race of this bearing was rigidly fixed to the shaft by a nut and lockwasher, and the outer race was fastened into the aluminum motor housing by pieces of the housing being bolted together.

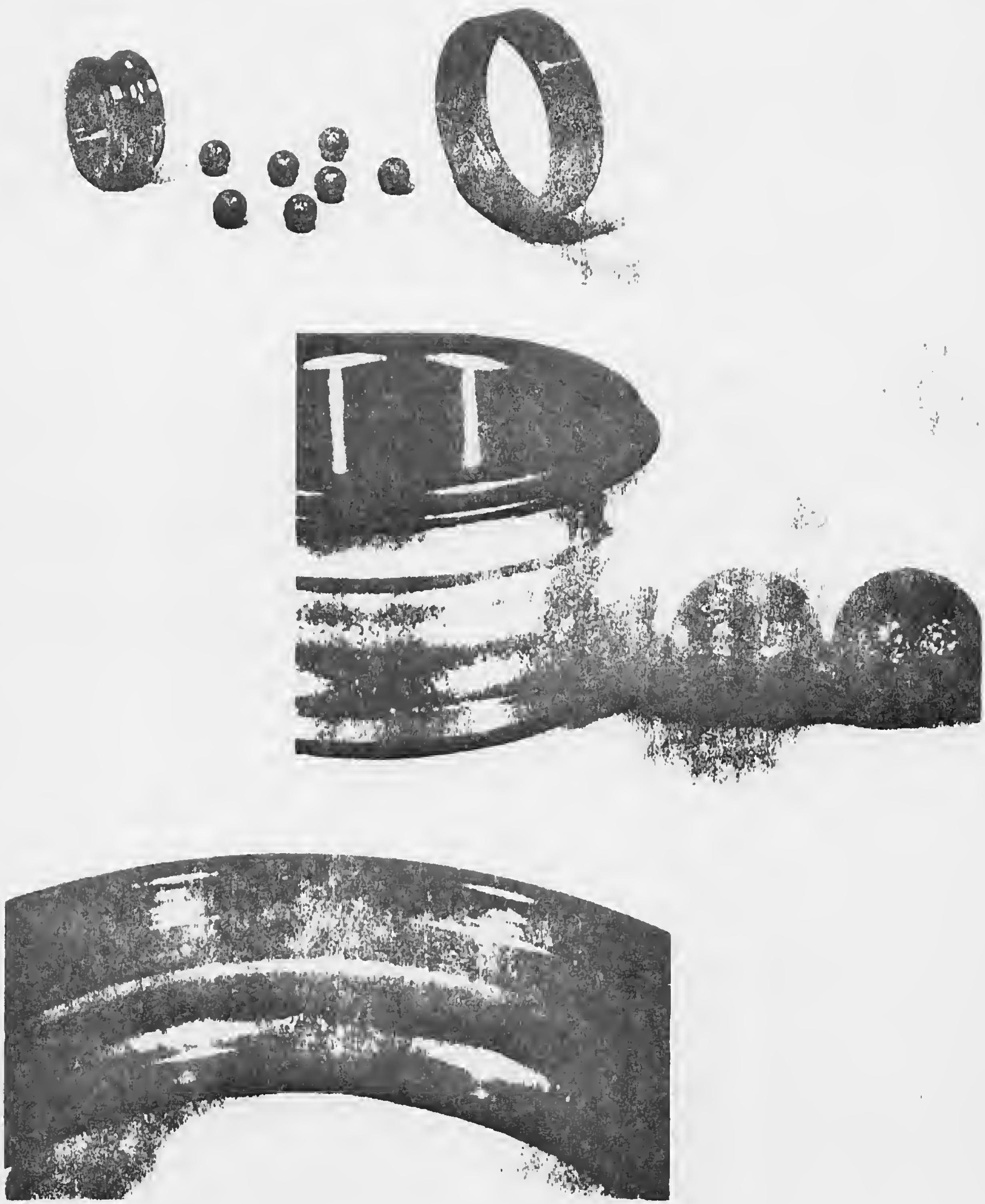
A small bearing, Barden No. 203SS5, was picked for the opposite end of the shaft. This rear bearing was not rigidly fixed to either shaft or housing, so it could move axially to allow for dimensional variations of manufacture and for expansion and contraction of motor parts with changes in temperature. A radial allowance of 0.0004 inches up to 0.001 inches was left between outer race and housing, and an allowance of zero up to 0.0003 inches between inner race and shaft on the rear bearing. To insure proper bearing geometry and to prevent the bearing from having its outer race turn in the housing, a pair of belleville springs were compressed with a force of 4 to 26 pounds between the rear of the housing and the outer race.

Although no detailed life calculations were made on the bearings, catalog data and design experience indicated to Task engineers that they would last well beyond the 2,500 hours required life, and that if failure occurred, it would first happen elsewhere in the motor. It was assumed that the greatest loads on the front bearing would be axial and less than 100 pounds, since the maximum pump preload spring force predicted by Geyser was 70 pounds. The Barden 204, it was felt, should be more than adequate for such loading. For the tail of the shaft, a smaller bearing was considered sufficient. There would be no need for the rear bearing to be large enough to slip over the spline, and the axial loading on it was expected to be very small, equivalent to the belleville spring load. Both bearings cost around $10. but by using the smaller bearing at the rear, there would be a few cents saved on each motor. Another benefit of the smaller bearing would be lighter weight. Excerpts from a Barden Company catalog giving specifications on the chosen bearings appear in Exhibit 8.

The first qualification test motor was made in the Task shop and tested on the Task dynamometer, without symptoms of difficulty. After this test, the motor was shipped, in late August, 1963, to The Geyser Pump Company where it was mated to a pump, installed in a test rig, and started on life tests according to the schedule of Exhibit 6. After 1,800 hours, Geyser technicians monitoring the test, noticed the motor was drawing excessive current. They shut the motor down and removed it from the pump. Turning the shaft over by hand, they could feel roughness in the bearings. (It wasn't possible to hear the bearings during test because the pump made too much noise.)

At 10 a.m. the next morning, several Geyser engineers appeared at Task with the motor asking for an answer to the problem. The motor was

Exhibit 1 - 204SST5 Bearing After 1800 Hours Operation

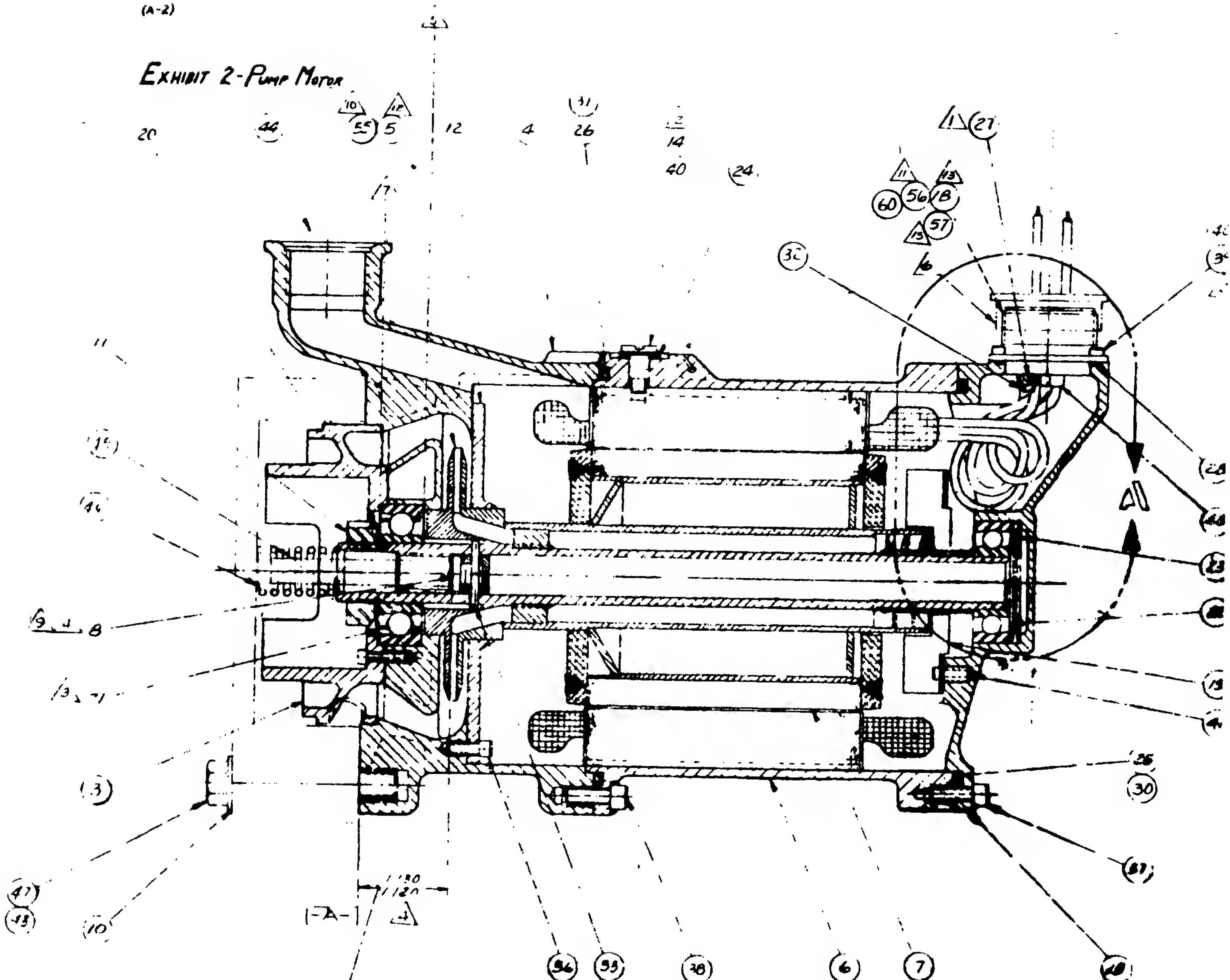


immediately disassembled in the Task shop. The front bearing had become severely galled, and the grooves of the races had been widened about 0.040 inches by wear all the way around. Wear was so extensive on both balls and races that the engineers could see no clues in the bearing as to how it had failed.

Meanwhile, the second qualification test motor had been finished and was ready for testing. Tooling had been completed and production was under way, with over a dozen motors complete and others in various stages of manufacture. Geyser Pump was progressing similarly in production of pumps to match the motors. Task top management had asked Mr. Jack Wireman, a mechanical engineer, to prescribe action for curing the problem.

(A-2)

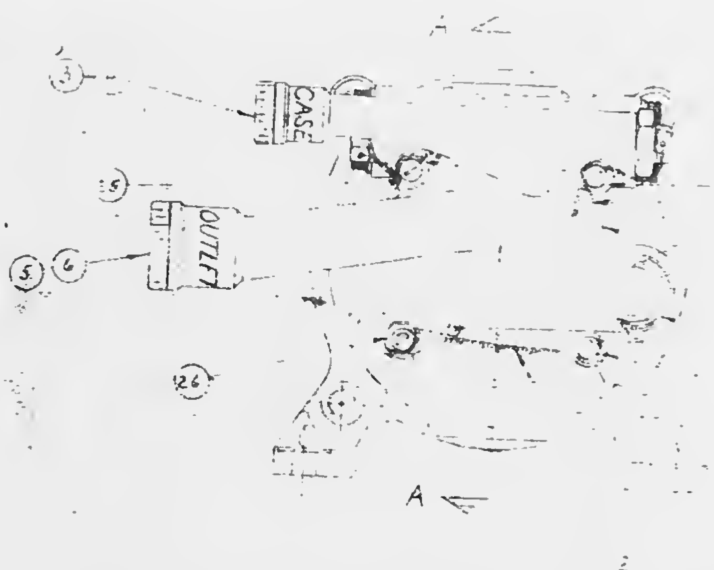
EXHIBIT 2-PUMP MOTOR

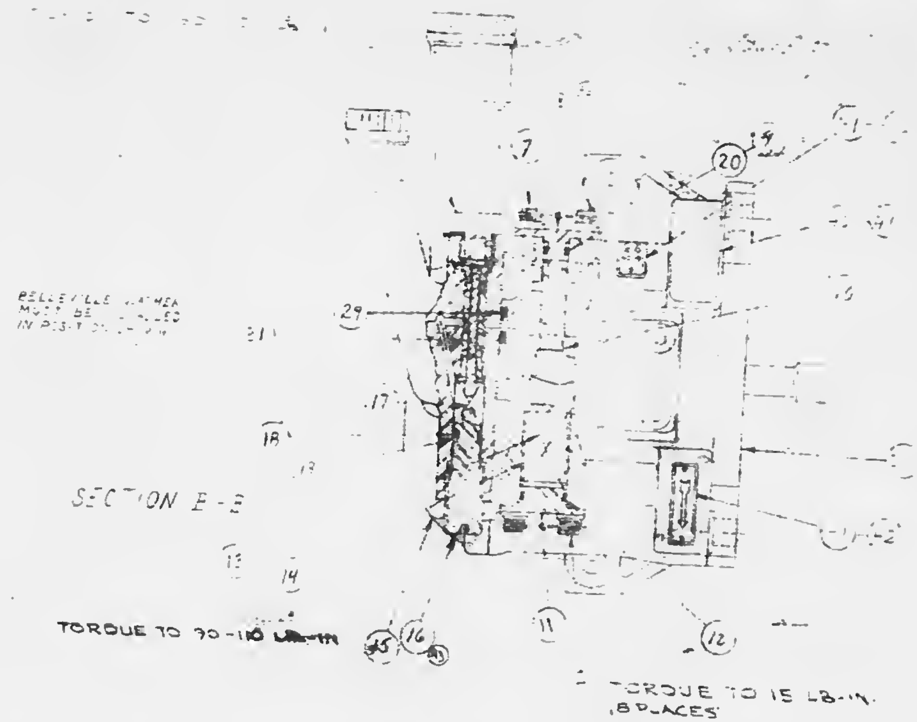


DIM TO BE HELD WHEN AXIAL LOAD OF 50 LB. MIN, 100 LB. MAX. IS APPLIED TO THRUST BUTTON
(MEASUREMENT OF THIS DIMENSION MAY BE OBTAINED BY VERTICALLY SUPPORTING THE ASSY OF THE PUMP HOUSING (ITEM 4) ROTOR ASSY (ITEM 7), IMPELLER (ITEM 5), BEARING (ITEM 21) & LOCKNUT (ITEM 11); SO THAT THE ROTOR WEIGHT IS SUPPORTED ONLY BY THE BEARING (ITEM 21) IN THE THRUST DIRECTION)

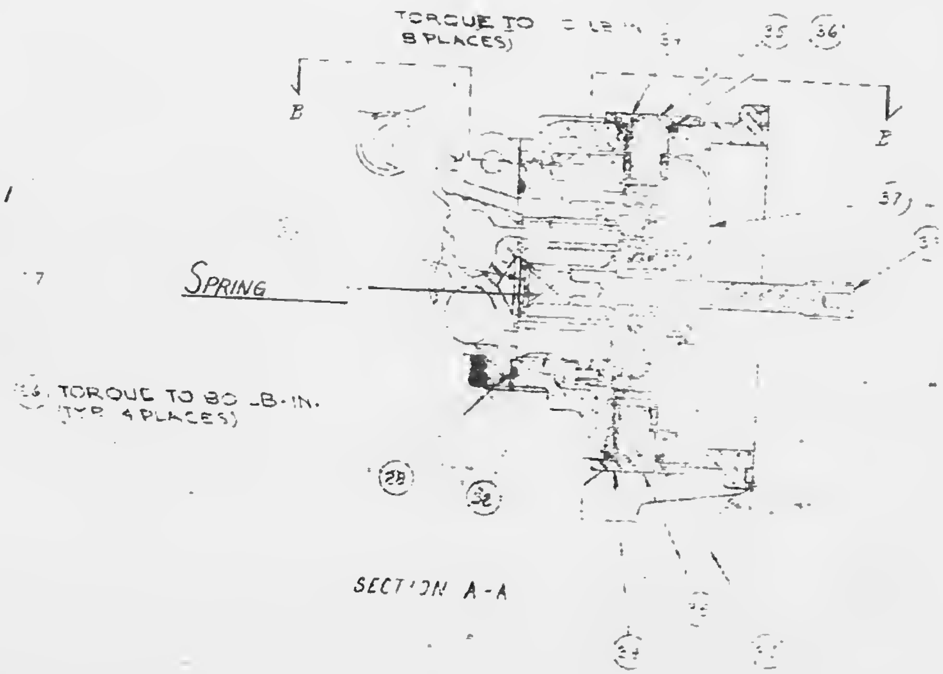
(A-3)

EXHIBIT 3 PUMP ASSY





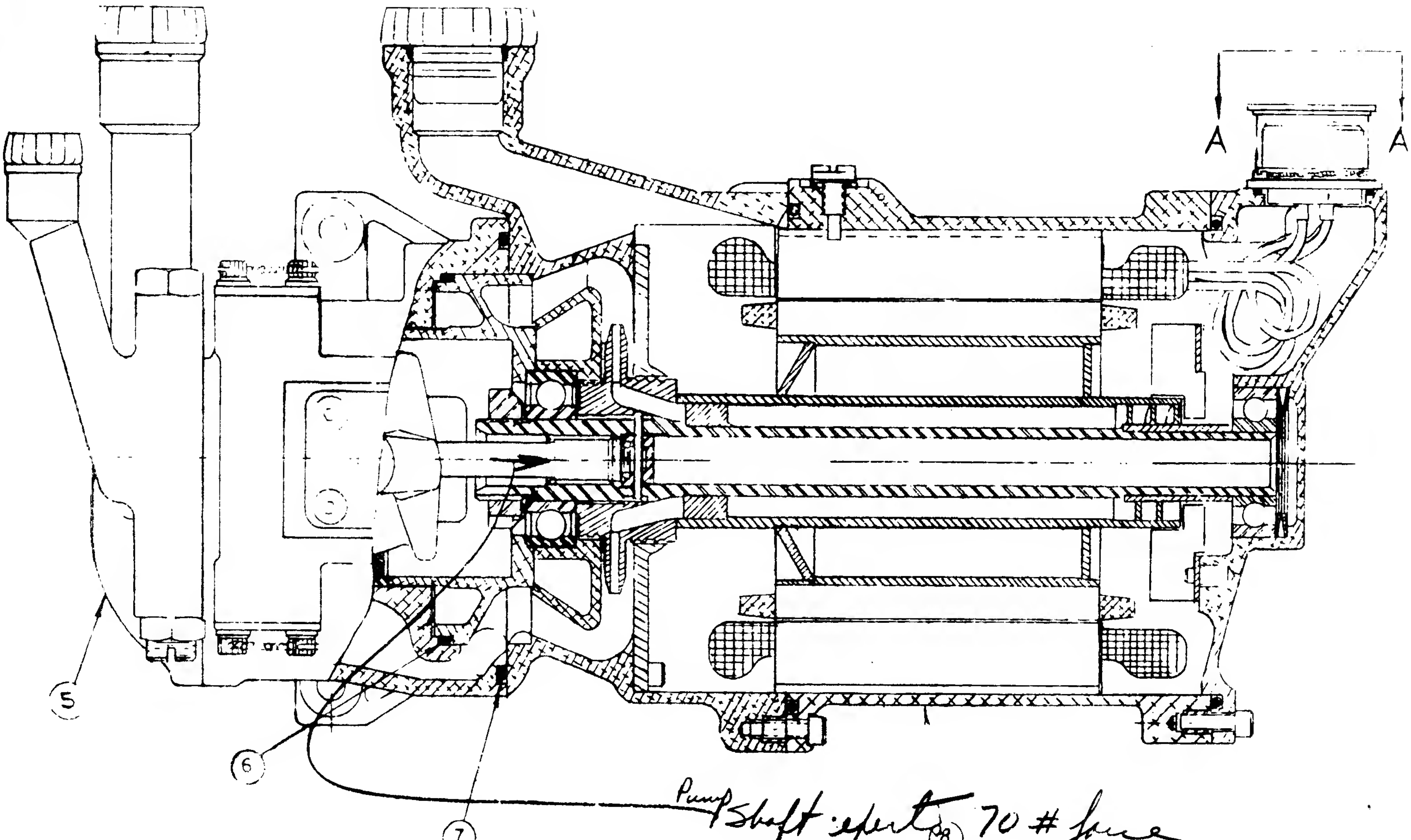
SECTION E-E



SECTION A-A

EXHIBIT 3 (CONT.)

(A-3)



Pump Shaft exerts (8) 70 # force
in direction shown
on plug in motor shaft.
(Picture slightly off.)

Exhibit 4 -

Properties

SKYDROL SPECIFICATIONS FOR NEW FLUIDS

Property	Skydrol 500A	Skydrol 7000
Appearance	Clear, purple liquid	Clear, green liquid
Neutralization number, mg. KOH/gm.	0.10 maximum	0.20 maximum
Specific gravity 25°C./25°C.	1.060 to 1.066	1.080 to 1.086
Viscosity, CS		
@ 210°F. (99°C.)	3.90 to 4.00	3.95 to 4.05
@ 100°F. (38°C.)	11.30 to 12.10	15.00 to 16.00
Moisture, per cent	0.20	0.25

TYPICAL PHYSICAL AND CHEMICAL PROPERTIES

Property	Skydrol 500A	Skydrol 7000
Appearance	Clear, purple liquid	Clear, green fluid
Odor	Mild, pleasant	Mild, pleasant
Autogenous ignition Temperature, (ASTM D 286-58T)	Above 1100° F. (593°C.)	1060°F. (571°C.)
Pour Point, (ASTM D97)	Below −85° F. (−65°C.)	Below −70°F. (−57°C.)
Neutralization number, (ASTM D-974-58T) mg. KOH/gm.	0.01	0.01
Specific gravity, (ASTM D941) (See Graph, page) 77°F. (25°C.)	1.065	1.086
Viscosity index	+238	+160
Viscosity, CS, (ASTM D445) (See Graph, page 14)		
210°F. (99°C.)	3.95	4.00
100°F. (38°C.)	11.70	15.50
Pressure Viscosity, CS 100°F., @ 4000 psi	20.8	Not Available
Thermal conductivity, 82°F. (28°C.)	0.0777 Btu/Hr./Ft.^{2}°F./Ft. (32.2×10^{-5}Cal/Sec. CM.^{2} °C. CM)	0.0723 Btu/Hr./Ft.^{2}°F./Ft. (29.9×10^{-5}Cal/Sec. CM.^{2} °C. CM)
Specific heat, @ 100°F., Btu/lb.,°F. or Cal/gm.°C.	0.39	0.42
Isothermal secant Bulk modulus 77°F., (See Graph, page 15)	340,000 psi	328,000 psi
Foaming (ASTM D892-46T)	Essentially nonfoaming	Essentially nonfoaming
Shear stability	Comparable to MIL-H-5606A	Exceeds MIL-H-5606A
Surface Tension, 77°F. (25°C.)	30.8 dynes/cm.	28.9 dynes/cm.
Hydrolytic stability	Skydrol 500A and Skydrol 7000 are not seriously affected by low concentrations of water (less than 2%), but water contamination should be avoided.	
Refractive index, n_D 25°C.	1.470 to 1.475	1.5067 to 1.507
Heat of Combustion	12,900 Btu/lb.	Not available
Thermal coefficient of expansion	0.000452/°F. 0.000813/°C.	0.000418/°F. 0.000753/°C.
Moisture %, (Karl Fischer)	0.05	0.05
Dielectric Strength (ASTM D877-49) 25°C.	12 KV	36 KV
Dielectric constant 25°C., 1 KC.	8.81	8.87
Volume resistivity (ASTM D-1169) OHM-CM 25°C.	43×10^{8}	500×10^{8}

Compressibility

Skydrol 500A and *Skydrol* 7000 both have a better bulk modulus than MIL-H-5606A, resulting in a more positive response of the hydraulic system. Table I shows the bulk modulus of *Skydrol* 500A, *Skydrol* 7000, water and MIL-H-5606A.

Table I ISOTHERMAL BULK MODULUS COMPARISON
77°F. (25°C.), 0 to 500 psi

Fluid	Bulk Modulus psi
Skydrol 500A	340,000
Skydrol 7000	328,000
Water	337,000
MIL-H-5606A	259,000

Volatility

The low volatility of *Skydrol* 500A and *Skydrol* 7000 assure low evaporation rates in the hydraulic system. Both fluids are relatively unaffected by changes in pressure and temperature; *Skydrol* 500A and *Skydrol* 7000 have far lower volatility than MIL-H-5606A or similar products. The Graph on page 16 shows vapor pressure vs. temperature for MIL-H-5606A, *Skydrol* 500A and *Skydrol* 7000.

Table II

Fluid	Viscosity Pressure Coefficient (VPC)*
Skydrol 500A	1.98
MIL-H-5606	2.52
MLO 7557	3.25
2-Ethyl Hexyl Sebacate	2.52

See SKYDROL 500A Viscosity vs. Pressure on page 14.

*Measured as per cent of change in viscosity per 100 psi pressure change.

Pressure Viscosity

The change of fluid viscosity with pressure will alter the fluid flow characteristics which could slow system response.

Skydrol fluids simplify this design engineering problem for pressure has relatively little effect on these fluids. The measure of the effect of pressure on viscosity, the Viscosity Pressure Coefficient, for *Skydrol* 500A is compared with other commonly used aircraft fluids in Table II.

Table III FOAMING TENDENCY OF SKYDROL 500A AND SKYDROL 7000 (ASTM D 892-46T)

	SKYDROL 500A		SKYDROL 7000	
	Tendency	Stability	Tendency	Stability
75°F. (24°C.)	40 ml.	22 sec.	5 ml.	2 sec.
200°F. (93°C.)	20 ml.	5 sec.	5 ml.	2 sec.
75°F. (24°C.)	40 ml.	19 sec.	5 ml.	2 sec.

Foaming Tendency

Both *Skydrol* 500A and *Skydrol* 7000 have a very low foaming tendency. The ability of these fluids to resist air "pick-up" and reject entrained air add greater reliability of the hydraulic system. Table III gives the sequence and results of foaming tests for both fluids.

Chemical Stability

Skydrol 500A and *Skydrol* 7000 are heat stable and resist oxidation at temperatures beyond those encountered in actual service. In hydraulic systems, the upper temperature limit for continuous operation for *Skydrol* 500A and *Skydrol* 7000 is approximately 225°F., (107°C.). Portions of the system can operate for a short time at higher temperatures without excessive deterioration of the fluids. At operational pressures of 3000 psi, *Skydrol* fluids have shown no tendency to thicken or form sludge over thousands of service hours.

Non-corrosiveness

Skydrol 500A and *Skydrol* 7000 are phosphate ester-based fluids which have little corrosive effects on the metallic rts of the hydraulic system—a fact proven by years flight experience. In general, the *Skydrol* fluids act as metal passivaters, which means they may be used as the preservative fluid in hydraulic components.

In mineral oil systems, a special preservative grade fluid is usually used to protect the component during storage. Systems equipped with the *Skydrol* fluids require no such special preservative grade fluid, the *Skydrol* fluids providing sufficient protection.

Table IV shows the low corrosive effects of both *Skydrol* fluids to common metals.

Table IV CORROSION AND OXIDATION TEST RESULTS
(per MIL H-5606, 168 hrs. at 250°F. (121°C.)

Property	Skydrol 500A Initial	Skydrol 500A Final	Skydrol 7000 Initial	Skydrol 7000 Final
Viscosity at 130°F. (54°C.) (cs.)	8.37	8.55	9.60	9.81
Neutralization number, mg KOH/gm.	0.02	0.03	0.13	0.28

Effect on Metals (weight change mg./cm.°)

Metal	Skydrol 500A	Skydrol 7000
Cu	−0.03	−0.92
Fe	−0.01	0.15
Al	0.00	0.02
Mg	−0.01	−0.12
Cd/Fe	−0.01	0.00
Fluid evaporation	<1.0%	2.05%
Fluid Separation	None	None
Color change	Fluid darkens slightly	Changes to darker green

at Transfer

Skydrol 500A and *Skydrol* 7000 have good heat transfer properties. Table V lists thermal data.

Table V THERMAL DATA OF SKYDROL 500A AND SKYDROL 7000

Property	Skydrol 500A	Skydrol 7000
Specific Heat		
90 to 120°F. (32 to 49°C.)	--	0.45 Btu/lb./°F.
−40°F. (−40°C.)	0.34 Btu/lb./°F.	--
75°F. (24°C.)	0.38 Btu/lb./°F.	--
145°F. (63°C.)	0.41 Btu/lb./°F.	--
212°F. (100°C.)	0.44 Btu/lb./°F.	--
Thermal Conductivity		
82°F., Btu/Hr.Ft.^{2}°F.Ft.	0.0777	0.0723
28°C., Cal/Sec.Cm.^{2}°C.CM	29.9 X 10^{-6}	32.2 X 10^{-5}
178°F., Btu/Hr.Ft.^{2}°F.Ft.	0.0779	0.0716
81°C., Cal/Sec. Cm^{2}°C.CM	32.2 X 10^{-5}	29.6 X 10^{-5}

Electrical Properties

Both *Skydrol* 500A and *Skydrol* 7000 possess good dielectric properties. If hydraulic system leaks develop, the *Skydrol* fluids do not cause short circuits, nor will they cause electrolytic corrosion in hydraulic systems. With the hundreds of miles of electrical wiring and the hundreds of circuits in today's aircraft, this lack of conductivity is another safety feature of *Skydrol* 500A and *Skydrol* 7000, which adds to their importance for hydraulic systems. Table VI shows the dielectric properties of the *Skydrol* fluids.

Table VI

Dielectric strength	Skydrol 500A	Skydrol 7000
25°C., 0.01" Gap	12 KV	36 KV
Dielectric Constant		
25°C., 100KC	8.81	8.87 (1 KC)
100°C., 100KC	6.95	
Power Factor		
25°C., 10KC	67%	0.49 (1 KC)
25°C., 100KC	6.7%	
100°C., 10KC	100%	
100°C., 100KC	39%	
Volume Resistivity, ohm-cm		
25°C., 500V., D.C., 0.1"Gap	43 x 10^{6}	500 x 10^{6}
100°C., 500V., D.C., 0.1"Gap	5.7 x 10^{6}	

Performance Properties

Fire Resistance

Skydrol 500A and *Skydrol* 7000 are much less susceptible to ignition than MIL-H-5606A, as shown by tests conducted according to AMS 3150. These tests were designed to evaluate the fluids under actual operating conditions to realistically pinpoint their fire-resistance value.

In specific tests, high-pressure sprays of *Skydrol* fluids through the white heat of a welder's torch (often above 6000°F.) does not cause burning, but only occasional flashing. In the same test, MIL-H-5606A ignites instantly and continues burning. On a red hot manifold at 1300°F., *Skydrol* fluids do not burn. In other tests simulating hot manifolds, sparks, exhaust flames or electrical arcing, *Skydrol* fluids do not support fire. Even though they might flash at exceedingly high temperatures, *Skydrol* fluids could not spread a fire because burning is localized at the source of heat. Once the heat source is removed or the fluid flows away from the source, no further flashing or burning can occur because of the self-extinguishing features of the *Skydrol* fluids. To those who know ignition sources these facts explain why *Skydrol* 500A and *Skydrol* 7000 protect the material investment in an aircraft and the lives of its passengers and crew.

Table VII. SHELL 4-BALL LUBRICITY TEST FOR SKYDROL 500A, SKYDROLL 7000 and MIL-H-5606A (600 RPM, 167°F., 1-hr., Scar Diameters in Millimeters)

Fluid	Steel On Steel 1 kg.	Steel On Steel 40 kg.	Steel On Bronze 1 kg.	Steel On Bronze 40 kg.
Skydrol 500A	0.154	0.809	0.327	0.894
Skydrol 7000	0.251	0.586	0.424	1.113
MIL-H-5606A	0.365	0.849	0.726	2.181

Lubricity

The experience of millions of flight hours on vi[illegible] every type of aircraft hydraulic pump have proven that *Skydrol* 7000 generally increases the service life of most

pumps. *Skydrol* 7000 in Douglas cabin superchargers is approved for 4000 hours of operation as compared to only [illegible]0 hours for petroleum oil. Experience with *Skydrol* 500A since its introduction in 1956 offers ample evidence that it too is an excellent lubricant. Tables VII and VIII show lubricity tests for both fluids.

Compatibility of Fluid Materials

Mineral Oil

Although *Skydrol* fluids are miscible with mineral oil, the mixing of *Skydrol* with mineral oil must be avoided to maintain *Skydrol* fire-resistant performance. Mineral oil will seriously degrade the ability of *Skydrol* to resist combustion and fire, the vital justification of equipping systems with *Skydrol*. Similarly, mineral oil, mixed with a *Skydrol* fluid, will degrade seals and packings used in *Skydrol*-equipped hydraulic systems.

Additives used in mineral oils can also damage *Skydrol* systems components; for example, some viscosity index (VI) improvers used with mineral oils may not be soluble *Skydrol*. In this case, the VI improver may precipitate from the hydraulic fluid mixture, leaving a gum-like residue which may interfere with proper operation of valves and filters.

Silicone and Silicate Fluids

Mixture of these fluids with a *Skydrol* fluid should be avoided for the same reasons mentioned for mineral oil.

Turbo Oil

Both *Skydrol* 500A and *Skydrol* 7000 are miscible with Turbo Oil 15 and 35 in concentrations up to 50 per cent; however, mixing of the fluids is not recommended, since such mixtures can degrade the fire-resistant properties of *Skydrol* and the packings and seals used in *Skydrol* equipped systems.

Miscibility of Skydrol 500A and Skydrol 7000

drol 500A and *Skydrol* 7000 are completely miscible, and the major effect of mixing the two fluids is to increase the viscosity of *Skydrol* 500A at low temperatures, as shown in Table X.

Table VIII VICKERS PF-3911 PISTON PUMP TEST

	Skydrol 500A		Skydrol 7000	
Duration Time:	224.2 hours		450 hours	
Lubricity:	Excellent. Superior to MIL-H-5606A		Excellent. Superior to MIL-H-5606A	
Packings:	Inspection of the Butyl elastomeric materials, including accumulator bladders, showed no degradation and practically no swelling.			
Viscosity (cs);	Initial	Final	Initial	Final
210°F. (99°C.)	3.37	2.26	3.92	2.95
100°F. (38°C.)	10.09	6.97	14.9	12.09
−40°F. (−40°C.)	576	407	6750	6130
Shear Loss:	20% in first 100 hours. Leveled off at approximately 33%.		17% in 400 hours	
Neutralization Number Change:	0.20 to 0.37		0.20 to 1.00	

Table IX COMPARISON OF SHEAR STABILITY
SKYDROL 500A, SKYDROL 7000 AND MIL-H-5606A

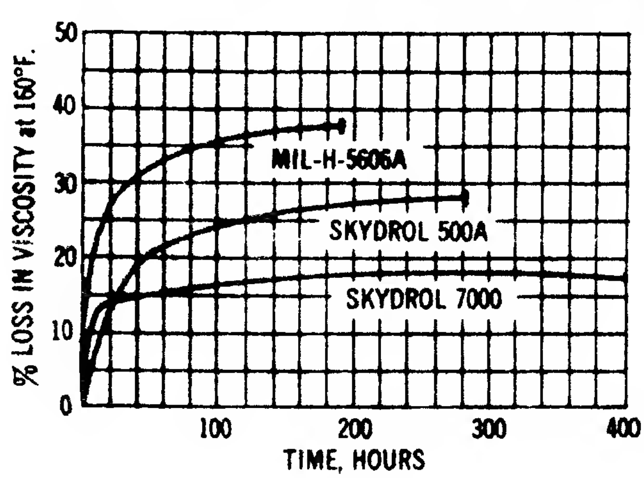


EXHIBIT 5

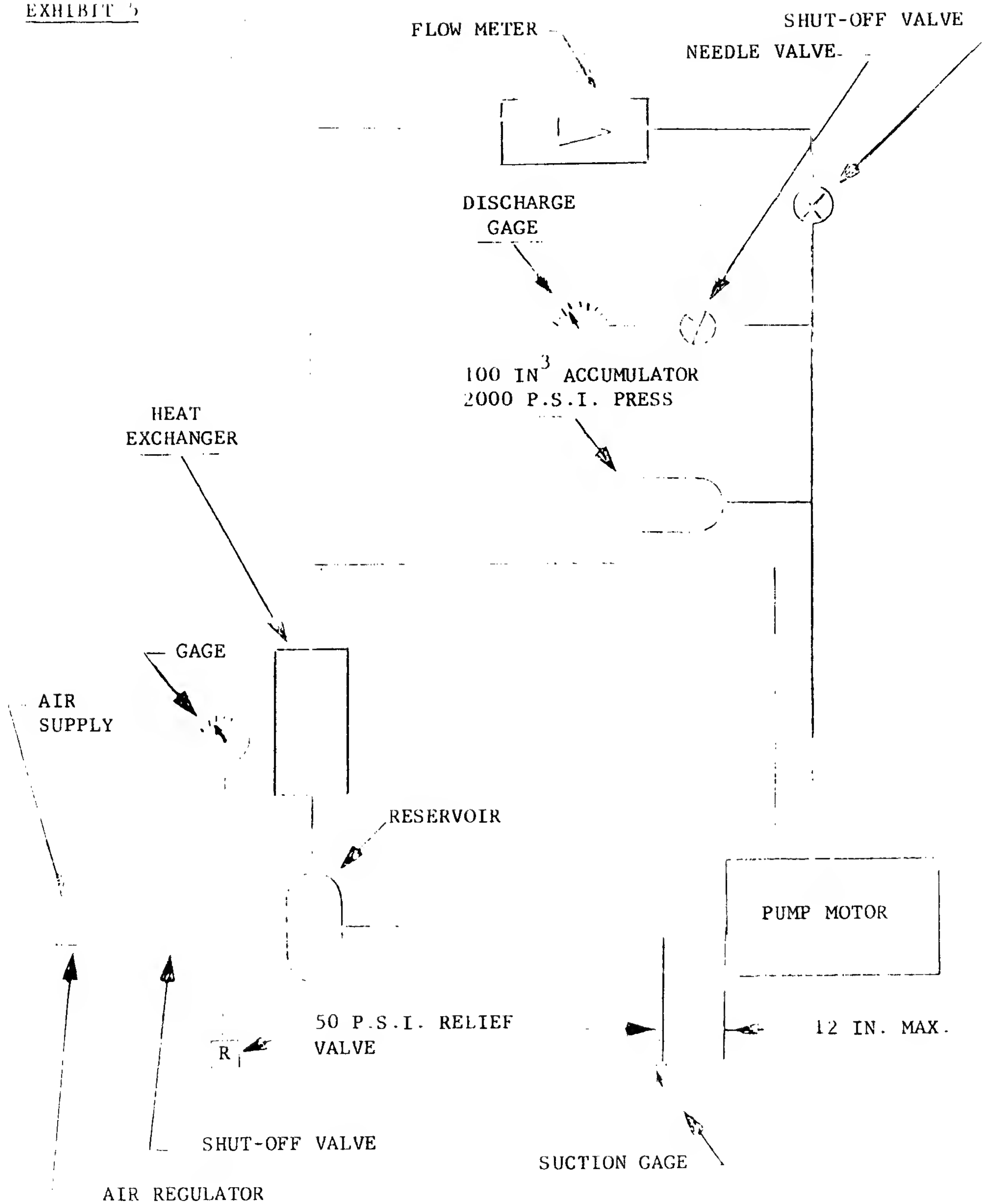


FIGURE 2. TEST SCHEMATIC

COPY

EXHIBIT 6 - Endurance Test Schedule

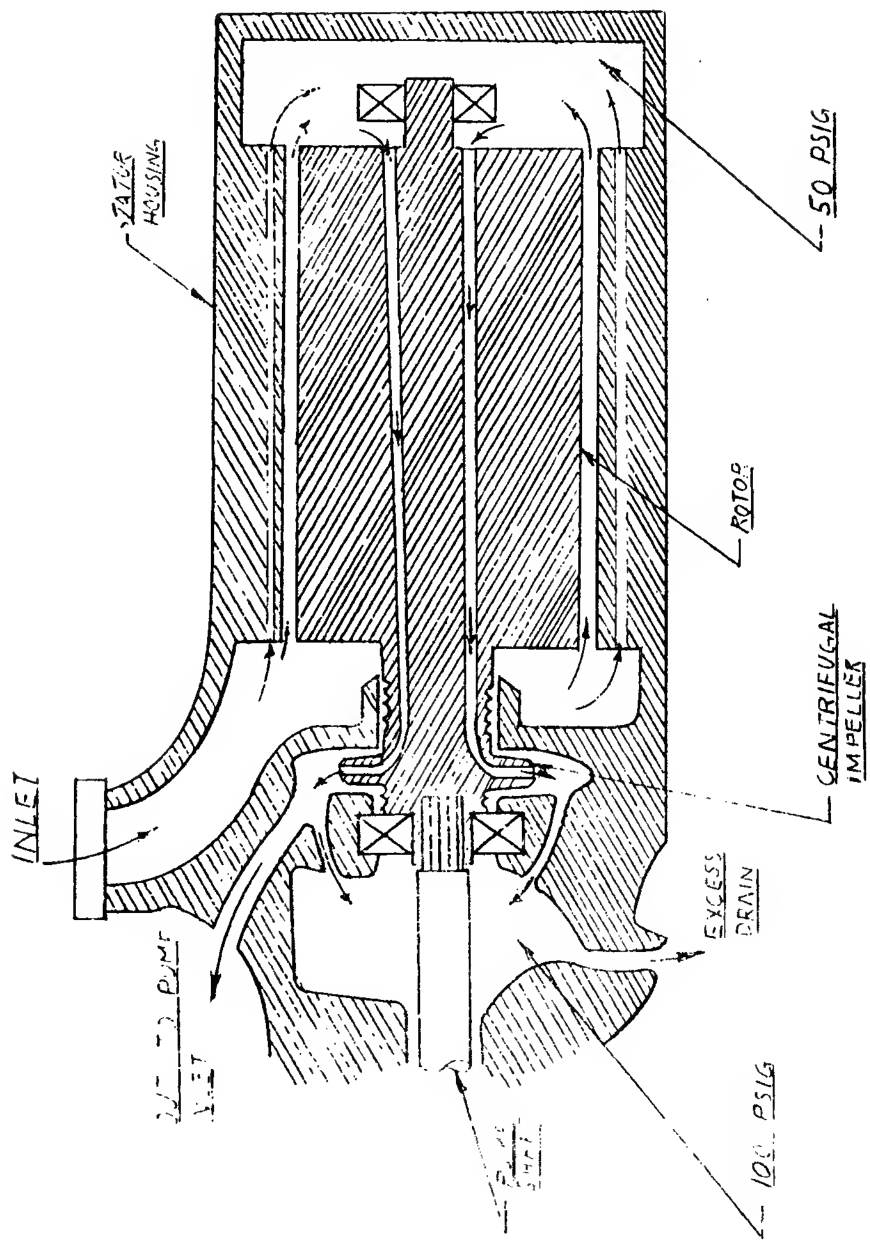
HOURS	OUTPUT	FLUID INLET PRESSURE (MIN.)	FLUID INLET TEMP. ± 5°F
225	Cycle A	5 psig	150°F
325	Cycle B	45 psig	150°F
110	1/2 gpm	45 psig	150°F
1820	1 gpm	45 psig	150°F
20	1 gpm	5 psig	Room Temp.

CYCLE A			CYCLE B		
Motor	Flow	Time	Motor	Flow	Time
On	5.7 gpm min.*	15 min.	On	5.7 gpm min.*	6.25 min.
On	0	30 min.	On	2 gpm	5 min.
Off	-	15 sec.	On	1/2 gpm	5 min.

*Flow which produces a motor current draw of 38.5 amperes or 5.7 gpm min., whichever occurs first.

ENDURANCE TEST SCHEDULE

Exhibit 7 - Schematic of Fluid Flow Through Motor



BARDEN STANDARD PRECISION BEARING TYPES

MATERIALS

Stainless steel (AISI 440C) rings and balls are used in most miniature and instrument bearings listed in this catalog. As heat treated and processed by Barden, these bearings have high corrosion resistance and good dimensional stability for high temperature operation. Average running torque values are the same as for chrome steel bearings.

Chrome steel (SAE 52100) rings and balls are used in most spindle and turbine bearings and special design bearings listed here. Chrome steel is preferred for many applications where corrosion resistance is not the governing factor.

Either material may be readily obtained in many standard bearings. Other materials for high temperature use, such as AISI M50 steel, can be supplied on special order.

CONSTRUCTION

Deep groove bearings have equal raceway shoulders on both sides of inner and outer rings, will support radial loads and thrust in either direction, and have lower torque at very low speeds than angular contact bearings.

Extra wide deep groove bearings, normally furnished with two shields, have increased lubricant space for long life at high speeds with the initial charge of lubricant.

Flanged deep groove bearings provide accurate positioning surfaces for good alignment; they permit through-boring, eliminating the need for housing shoulders or shoulder rings.

Angular contact bearings have one shoulder cut away on either inner or outer ring, will support radial loads and thrust in one direction only, and require light axial loading; they are generally preferred for the higher speed applications.

Separable angular contact bearings have one shoulder eliminated on the inner ring. This permits removal and mounting of ring on shaft while the outer ring assembly is mounted separately in housing, an aid to assembly and dynamic balancing.

Non-separable angular contact bearings have one shoulder only partially cut away on the outer ring to prevent separation of bearing parts. Construction allows use of one-piece retainer and more balls for greater static and dynamic load capacity than deep groove bearings.

BALL RETAINERS deep groove bearings

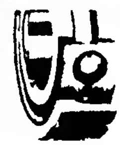
Pressed steel standard retainers, used for moderate to high speeds, are one-piece in smaller sizes, two-piece in larger. Two-piece "W" retainers, also available in most sizes, have "antiwindup" design to prevent retainer lock, reduce torque peaks and increase bearing life at speed. Retainers are relatively unaffected by temperature; lubricant limitations govern operating temperatures.

...nolic retainers, used for high speeds, are one-piece "TA" in smaller sizes, two-piece "T" in larger; both are outer ring piloted for increased lubricant penetration and circulation, less wear and longer life. "T" retainers have aluminum reinforcement and positive riveting to strengthen lightweight, long wearing phenolic. Retainers operate up to about 300°F (150°C) and for short periods to about 350°F (175°C).

BALL RETAINERS angular contact bearings

Phenolic retainers for separable bearings are one-piece, outer ring piloted for best endurance at high speeds. Will operate to about 300°F (150°C) and for short periods to about 350°F (175°C).

Phenolic retainers for non-separable bearings are one-piece, lightweight Barden "halo" type, outer ring piloted for better lubricant penetration and circulation, longer life at high speeds. Will operate to about 300°F (150°C) and for short periods to about 350°F (175°C).



Bronze retainers for non-separable bearings, used for high speeds, have high conductivity to dissipate heat, are one-piece, thin section and outer ring piloted for increased lubricant penetration, circulation and cooling, resulting in longer bearing life at high speeds and temperatures. Bronze retainers are relatively unaffected by temperatures up to 550°F (285°C); range of operating temperatures is largely governed by lubricant limitations.

SHIELDS AND SEALS

Stainless steel shields, single or double, are available in most sizes of deep groove bearings. Barden close-clearance shield design protects against contamination and lubricant loss; precision snap wire shield retention avoids outer ring distortion. Single-shield flanged bearings have shields on flange side. Shields and snap wires are removable.

Flexeal seals, exclusive with Barden, are low friction, wear resistant fiber bonded to aluminum; in light contact with ground surfaces on inner rings, Flexeals have positive but free sliding action that effectively seals in lubricant, seals out contaminants and minimizes air or fluid flow through bearings. Precision snap wire retention avoids outer ring distortion. Available in many deep groove sizes.

DUPLEX MATCHED PAIRS

Barden-matched duplex pairs are available in most deep groove and angular contact non-separable bearings listed in this catalog. For further data see page 52.

Matched preloaded pairs, either DB (back to back) or DF (face to face), support radial loads and thrust in either direction, they provide exact axial positioning, increased rigidity, controlled axial and radial yield rates, and increased radial load capacity.

Matched tandem pairs, DT (back to face), support radial loads and thrust in one direction only; they should be mounted opposed with thrust loading between pairs or between a pair and a single bearing. Used mainly for increased load capacity and rigidity.

NOMENCLATURE

STANDARD BEARING TYPES. Standard types of Barden Precision bearings can be identified by use of the following nomenclature which is used on order acknowledgments, packages, invoices and correspondence. For new designs or initial orders, Barden should be furnished complete environmental and performance requirements so that bearings best suited for the application may be specified. Well established bearing requirements may be specified precisely by using the complete nomenclature shown here.

MATERIAL for rings and balls.
- S=AISI 440C stainless steel (except A500 sizes). Omit S for SAE 52100 chrome steel.
- A=AISI 440C stainless steel (A500 sizes only).

BEARING TYPE, if other than deep groove.
- B=separable angular contact.
- H=non-separable angular contact.

BALL RETAINER, if other than standard pressed steel in deep groove bearings or one-piece phenolic in angular contact.
- W=two-piece, thin-section "antiwindup" retainer.
- TA=one-piece snap phenolic.
- T=two-piece aluminum reinforced phenolic
- JB one-piece bronze "halo" type.
- Z=spring seperators.

RADIAL PLAY CODE, as given on bearing listing pages.

TORQUE TESTED miniature or instrument bearing.
- VL=torque tested.
- V=torque controlled.

CALIBRATION.
- C=standard calibration on bore and O.D.
- CXO=calibration on bore only.
- COX=calibration on O.D. only.

See page 72 for calibration codes.

SERIES AND SIZE.
- F=flanged outer ring.
- W=extended inner ring.

Other letters and numerals identify bearing series and basic size.
- K=separating symbol used if radial play code number would otherwise follow a numeral.

SHIELDS AND SEALS.
- S=single shield.
- SS=double shield.
- F=single Flexeal seal.
- FF=double Flexeal seal.

SPECIAL FEATURES.
- X followed by a number = deviations from standard, usually bore, O.D. or width dimensions.
- K=separating symbol used if radial play code number would otherwise follow a numeral.

DUPLEX MATCHED PAIRS.
- DB=back to back.
- DF=face to face.
- DT=tandem.
- D=universal match, DB, DF or DT.

Numeral, if any, following is special preload or matching load in pounds.

LOW RADIAL RUNOUT.
- E=.000050" maximum inner ring runout.

Number, if any, following indicates other specific controls of maximum runout.
- R=high point of eccentricity marked on inner ring.
- R1=high point marked on outer ring.
- R2=high points marked on both rings.

LUBRICANT.
- O=oil.
- G=grease.
- SG=preservative compound.
- V=vacuum impregnation.

See listing pages for lubricant normally supplied. See page 67 for lubricant codes most frequently used for specific needs.

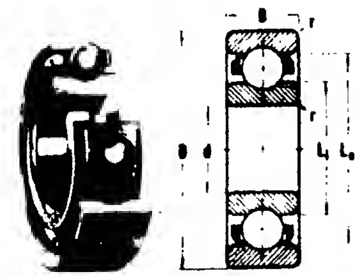
	Material	Series and Size	Bearing Type	Shields and Seals	Ball Retainer	Special Features	Radial Play Code	Duplex Matched Pairs	Torque Tested	Low Radial Runout	Calibration	Lubricant
EXAMPLE NO. 1:	S	R144		SS	W	X3K	3		V		C	O-11
EXAMPLE NO. 2:		102	H				5	DB5		E		G-2

SPECIAL DESIGN BEARINGS AND ASSEMBLIES. Barden bearings or assemblies designed specifically to customer requirements are identified by Y or Z followed by a number, such as Z155. This basic nomenclature generally includes all specifications normally spelled out by additional nomenclature in the case of standard bearing types. Occasionally, later modifications will be reflected in the bearing nomenclature by the use of X followed by a number, such as the Z202X1.

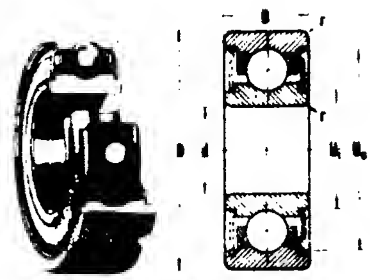
BARDEN PRECISION SPINDLE AND TURBINE BEARINGS

LOW TO ULTRA HIGH SPEED

DEEP GROOVE
STEEL OR PHENOLIC RETAINERS



OPEN



SHIELDED

AVAILABILITY. All sizes listed are normally available from stock in chrome steel.

DIMENSIONS

BORE d mm	BORE d inches	O.D. D mm	O.D. D inches	WIDTH B mm	WIDTH B inches	MAX FILLET RADIUS inches r	STEEL RETAINER Open	STEEL RETAINER Single shield	STEEL RETAINER Double shield	PHENOLIC RETAINER Open	PHENOLIC RETAINER Single shield	PHENOLIC RETAINER Double shield	OTHER DIMENSIONS inches L_i	L_o	U_i	U_o	APPROX. WEIGHT pounds	DATA REFERENCE NUMBER
15	5906	32	1.2598	9	.3543	012				102T			.798	1.053			.07	102
15	.5906	35	1.3780	11	.4331	.025	202K	202S	202SS	202T	202ST	202SST	815	1.153	.755	1.223	.10	202
15	5906	35	1.3780	12.70	.5000	.025					202STX1	202SSTX1			.755	1.223	.20	202
17	6693	35	1.3780	10	.3937	.012	103K	103S	103SS	103T	103ST	103SST	895	1.153	.835	1.215	.09	103
17	.6693	40	1.5748	12	.4724	.025	203K	203S	203SS	203T	203ST	203SST	.952	1.292	.890	1.372	.20	203
20	.7874	42	1.6535	12	.4724	.025				104T	104ST	104SST	1.050	1.390	.989	1.458	.18	104
[illegible]	7874	47	1.8504	14	.5512	.040				204T	204ST	204SST	1.130	1.530	1.060	1.610	.20	204
	9843	47	1.8504	12	.4724	.025				105T			1.247	1.587			.21	105
25	9843	52	2.0472	15	.5906	.040	205K	205S	205SS	205T	205ST	205SST	1.320	1.720	1.250	1.800	.30	205
30	1.1811	55	2.1654	13	.5118	.040				106T	106ST	106SST	1.511	1.869	1.451	1.949	.30	106
30	1.1811	62	2.4409	16	.6299	.040				206T	206ST	206SST	1.580	2.060	1.500	2.200	.50	206
35	1 3780	62	2.4409	14	.5512	.040	107K	107S	107SS	107T	107ST	107SST	1.710	2.110	1.620	2.190	.35	107
35	1 3780	72	2.8346	17	.6693	.040				207T	207ST	207SST	1.857	2.382	1.777	2.523	.70	207
40	1.5748	80	3.1496	18	.7087	.040				208T			2 081	2.643			.80	208
45	1.7717	85	3.3465	19	.7480	.040				209T			2.289	2.850			.90	209

APPLICATIONS. Motors, generators, aircraft accessories, gear drives, pumps, power tools, compressors, machine tool spindles, stable platforms, magnetic recording devices and other low to ultra high speed applications.

DESCRIPTION. Metric series deep groove bearings for smooth, quiet operation with minimum vibration under moderate to heavy loads; support radial loads and thrust in either direction. Some sizes (suffix X) are extra wide to hold more lubricant and [illegible]e wider mounting surfaces for accurate alignment.

[illegible]all retainers for low to high speeds are two-piece pressed steel; for ultra high speeds, two-piece aluminum-clad phenolic laminate, outer ring piloted for increased lubricant penetration and circulation, less wear and longer life. Aluminum reinforcement and positive riveting add strength to light weight and long wearing qualities of phenolic.

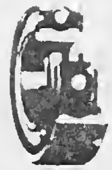
Close clearance shields protect against contamination and lubricant loss; precision snap wire shield retention avoids outer ring distortion. Shields and snap wir[illegible] are removable.

MATERIAL. **Rings and balls:** SAE 52100 chrome bearing steel. **Retainers:** pressed stainless steel or aluminum reinforced phenolic. **Shields and snap wires.** stainless steel. Further data on page 48.

LUBRICANTS NORMALLY SUPPLIED. Oil: MIL-L-6085A (Barden code 0-11) or MIL-L-7808C (Barden code 0-14) for open bearings only; both require subsequent lubrication wth continuous oil mist or spray application. **Grease:** Andok C (Barden code G-6) or UniTemp 500 (Barden code G-18). **Preservative compound:** MIL-C-11796A (Barden code SG-1) for open bearings only; requires subsequent lubrication with mineral oil. Further data on page 66.

PRELOADED PAIRS. Most sizes listed may be obtained in DB or DF duplex pairs with controlled axial preload. Further data on page 52.

FURTHER DATA. Tolerances, page 46. Radial and axial play and yield, page 49. Shaft and housing shoulders and fits, page 68.



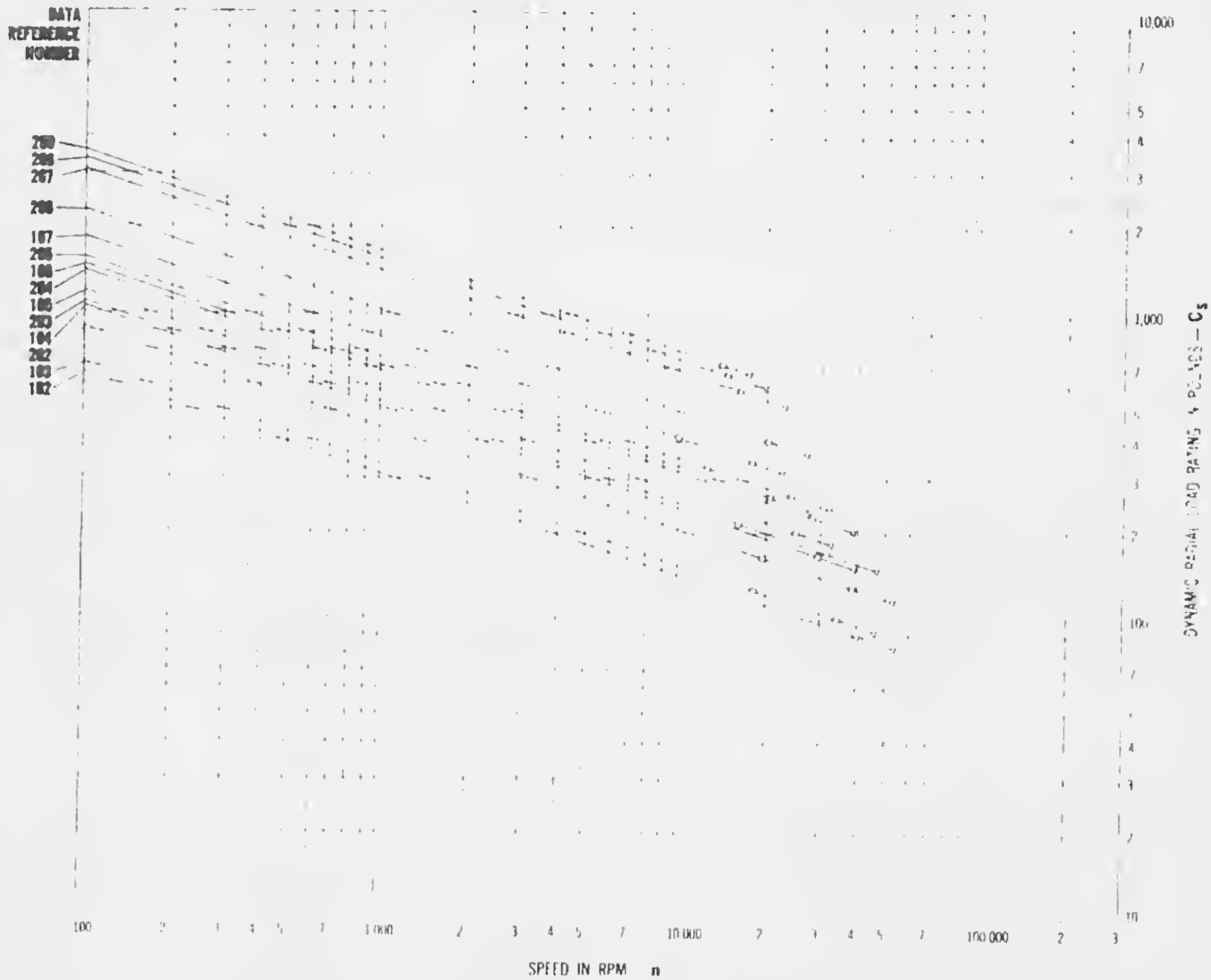
PERFORMANCE DATA

DYNAMIC LOAD RATINGS. Radial load ratings below are for 500-hour design life, or 2500-hour average life, with proper mounting, lubrication and protection of bearings. Use value C_s for fatigue life computation procedure on page 62.

○ Normal speed limit for open or shielded bearings with steel retainers.

□ Normal speed limit for open or shielded bearings with phenolic retainers, grease lubrication.

△ Normal speed limit for open bearings with phenolic retainers, oil mist or spray lubrication.



DATA REFERENCE NUMBER	STATIC LOAD RATINGS—pounds THRUST LOAD—T_s RADIAL PLAY RANGE Code 3 .0002"-.0004"	Code 5 .0005"-.0008"	Code 6 .0008"-.0011"	RADIAL LOAD C_s
102	2030	2120	1800	650
103	2300	2350	1700	730
	.0002"-.0006"	.0005"-.0009"	.0009"-.0014"	
202	2100	2240	2300	770
203	2720	2880	3000	1000
104	2160	2210	2340	900
204	3900	4000	4160	1400
105	[illegible]	[illegible]	2400	950
[illegible]	[illegible]	4650	4680	1800
106	[illegible]	3920	3630	1300
[illegible]	[illegible]	[illegible]	[illegible]	1700
		.0009"-.0012"	.0012"-.0017"	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	3100
[illegible]	[illegible]	[illegible]	10400	5000
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

STATIC LOAD RATINGS. Static ratings are shown at left for radial play ranges normally supplied. These values indicate peak loads that can be sustained by bearings without permanent effect on smoothness of operation. Further data on page 60.

AVERAGE RUNNING TORQUE. These bearings are used mainly at speeds where the predominant factor in torque is lubricant drag, which varies with lubricant type and quantity, speed and operating temperatures. When needed for estimate of power requirement, approximate torque values will be supplied.

LOAD CAPACITY

In many applications final bearing selection can be made from dimensional and performance data shown on the bearing listing pages of this catalog. Where an application runs continuously at speed, or where static load capacity is a major factor, bearing selection can be finalized only by computation of life and load factors.

Fatigue life of ball bearings that run continuously at speed under appreciable loading is considered to be the number of hours or revolutions that the bearings run before the first evidence of ball or raceway spalling develops. Fatigue is directly related to bearing load and speed. Calculation of fatigue life is of primary importance when selecting bearings for use in power-driven devices sush as motors, generators, gear drives, turbines and similar equipment.

Bearing life may be limited by factors other than fatigue life, such as lubricant exhaustion, contamination misalignment or improper fitting, and thermal constraint. In some cases, static capacity may be more important than fatigue life in selection of a bearing, as in lightly loaded components such as gyro gimbals, synchros and computer gear trains where little sustained speed is involved.

STATIC LOAD RATINGS

The static load capacity of a bearing is the peak load that can be sustained without appreciable permanent effect on smoothness of operation. Static load ratings shown on the listing pages of this catalog are based on load rating evaluation methods developed by the Anti-Friction Bearing Manufacturers Association. These ratings may be exceeded for some applications, but users anticipating heavier loadings should consult Barden before finalizing selection

In a lightly loaded ball bearing at rest, small elastic deformations are developed at ball-to raceway contacts. As loads increase, a portion of these deformations become permanent and, if sufficiently large, impair subsequent performance by causing high torque or rough operation. For most precision applications, a reasonable limit value for permanent indentations is one ten-thousandth of the ball diameter.

Under pure radial loading, the maximum depth of indentations approaches the center of the raceway. Under thrust loading, the indentations approach the edge of the raceway. With increased radial play, the contacts approach still closer to the edge of the raceway. Under heavy thrust loads and large values of radial play, the contact areas may run over the edge of the raceway, resulting in increased stress and deeper indentation. These effects, along with raceway depth limitations, have been considered in establishing the static thrust load ratings in this catalog.

In general, less damage is done to bearings if loads are imposed while the bearings are rotating than when at rest. Even in high speed operation, however, if the load duration is so short that there is no rotational overlap of the most heavily stressed ball contact areas, separate permanent indentations may result. Bearings used in high speed applications where smooth operation is important, such as gyro rotors, should be selected so that maximum peak or shock loads are within static ratings.

In many cases, a simple statement of g loading is not sufficient for use in bearing selection. Transient and vibratory loads must also be considered, but they are frequently difficult to determine, since resonance and damping characteristics of the complete rotating assembly and supports will greatly affect peak loadings imposed on the bearings. Complete vibration analysis or qualification tests under actual vibratory conditions may be needed.

BEARING SELECTION FOR STATIC CAPACITY

Pure radial loads on the bearing tentatively selected can be checked directly against the radial load rating in column C_o of the static load table on the appropriate listing page. Pure thrust loads can be checked against the thrust load columns T_o under the radial play range selected. Combined radial and thrust loads must be computed in terms of their radial and thrust load components, with bearing selection made so that each of the component loads is within its respective static radial or thrust rating. If either radial or thrust loading rating is lower than the peak load expected, a bearing with higher static rating should be chosen.

Static load ratings for DT (tandem) duplex pairs are double the thrust and radial ratings of single bearings. With usual preload, the thrust rating of a DB (back to back) or DF (face to face) pair is equal to that of a single bearing but radial capacity is double that of the single bearing.

SPEED CAPABILITIES

Speed capability of bearings is almost impossible to determine exactly because of the wide variety of environmental, design and life requirements. However, the approximate speed limits given in the dynamic load rating charts on the listing pages of this catalog will serve for general guidance. On request, Barden engineers will assist you in making detailed studies of specific speed and life requirements as an aid in bearing selection

Bearings with phenolic or bronze ball retainers have highest speed capability; bearings with pressed steel retainers have lowest speed capability. Speeds for bearings with phenolic or bronze retainers are limited by type of lubrication or by outer ring centrifugal ball loading; speeds for bearings with steel retainers are limited by retainer performance.

Bearings with phenolic or bronze retainers have highest speed-life capability when lubrication is supplied during operation by continuous oil spray or mist. Prelubrication with grease gives lower speed-life capability. Lowest speed-life capability results when bearings are prelubricated with instrument oil and not supplied with additional lubricant during operation

FATIGUE LIFE

For bearings that are properly mounted, lubricated and protected, fatigue life may be estimated as the number of hours or revolutions at a given speed and load that a group of similar bearings will operate before the first evidence of spalling or flaking of raceways or balls become apparent.

Since some variability in fatigue life is inevitable, it is necessary to establish a probability factor for the percentage of a group of bearings

hat will endure the given conditions of load and speed. In accordance with bearing industry practice, a probability factor of 90% is employed for load-life computations in this catalog.

Design fatigue life is considered to be the estimated life which will be exceeded by 90% of a group of identical bearings operating under identical load conditions, assuming proper mounting, lubrication and protection against foreign substances. Average fatigue life is approximately five times this figure.

Moderate changes in the load applied to the bearing have a pronounced effect on fatigue life. Halving the bearing load increases life eight times, but doubling the load reduces life to one-eighth. For this reason, an accurate determination of actual operating loads is most important for life computations.

BEARING SELECTION FOR DYNAMIC CAPACITY

Equivalent radial load computations

Fatigue life computations are based on two assumptions: a constant direction radial load and a condition of stationary housing (outer ring) and rotating shaft (inner ring). Therefore, combined radial and thrust loads must be converted into an equivalent radial load. Also, even where there is zero thrust load, an equivalent radial load must be calculated to take account of any radial load that tends to concentrate on a small ortion of the inner ring raceway. Such an effect is developed by dynamic unbalance loading on a rotating inner ring or by a dead weight or constant direction load on a stationary inner ring. The formulas below for equivalent radial loads take these factors into consideration.

Equivalent radial loads for individually mounted bearings are found by solving both Formulas 1 and 2 below. The larger of the two values derived is then used in subsequent formulas to find fatigue life.

FORMULA 1 $P = R_h + 1.2R_n$

FORMULA 2 $P = X(R_h + 1.2R_n) + YT$

where:
P = equivalent radial load, pounds
R_h = radial load fixed in relation to outer ring, pounds
R_n = radial load fixed in relation to inner ring, pounds
X = radial load factor
Y = thrust load factor
T = thrust load, pounds

Identification of radial loads R_h and R_n is found in Table 7.

Factor X is found by using the DATA REFERENCE NUMBER of the bearing tentatively selected to enter Table 8 (pages 64 and 65) to find the contact gle for the radial play range chosen. This contact angle is then entered in Chart A (page 62) to determine factor X.

When appreciable thrust loads are involved, select the highest possible radial play range for the highest resulting contact angle, since both factors X and Y decrease as radial play is increased.

To find factor Y first note the appropriate value ZD^2 in Table and compute value $\frac{T}{ZD^2}$. Select Chart B or Chart C (page 62) as shown by Table 8 and enter with $\frac{T}{ZD^2}$ and the contact angle of the bearing to determine factor Y.

When Y has been established, Formulas 1 and 2 may be computed and the larger resulting value taken to Formula 3 to find the design fatigue life of the bearing selected.

TABLE 7—RADIAL LOAD SYMBOLS R_h AND R_n

Component rotation	Nature of radial load	Symbol
Housing (outer ring) stationary, shaft (inner ring) rotating	Dead weight or constant direction load fixed in relation to housing (outer ring)	R_h
	Dynamic unbalance or rotational load fixed in relation to shaft (inner ring)	R_n
Shaft (inner ring) stationary, housing (outer ring) rotating	Dead weight or constant direction load fixed in relation to shaft (inner ring)	R_n
	Dynamic unbalance or rotational load fixed in relation to housing (outer ring)	R_h

EXAMPLE—EQUIVALENT RADIAL LOAD COMPUTATION

Application	high speed turbine
Operating speed	80,000 rpm
Rotating member	shaft (inner ring)
Lubrication	oil spray or mist
Dead weight radial load	4.3 pounds
Dynamic unbalance at operating speed	2.3 pounds
Thrust from turbine	15.0 pounds
Thrust from preload spring	8.0 pounds
Bearing tentatively chosen	38H
DATA REFERENCE NUMBER	38H

From Table 7:
Dead weight radial load of 4.3 pounds = R_h
Dynamic unbalance of 2.3 pounds = R_n
Total thrust $T = 15.0 + 8.0 = 23.0$ pounds

Using Formula 1: $P = R_h + 1.2R_n = 4.3 + 1.2(2.3) = 7.1$ pounds

To obtain values X and Y for Formula 2, enter Table 8 with DATA REFERENCE NUMBER 38H to find the contact angle of the bearing. The code 6 radial play range shows a contact angle of 17° as compared with 14 for code 5. Selecting code 6 for the higher contact angle and entering Chart A with 17°, factor X = .43. Entering Table 8 with reference 38H, $ZD^2 = .220$

Value $\frac{T}{ZD^2} = \frac{23}{.220} = 105$

CHART A—RADIAL LOAD FACTOR X

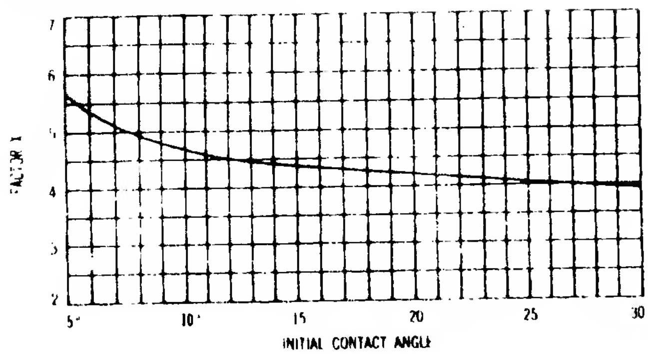


CHART B—THRUST LOAD FACTOR Y OR Y_d (SEE TABLE 8)

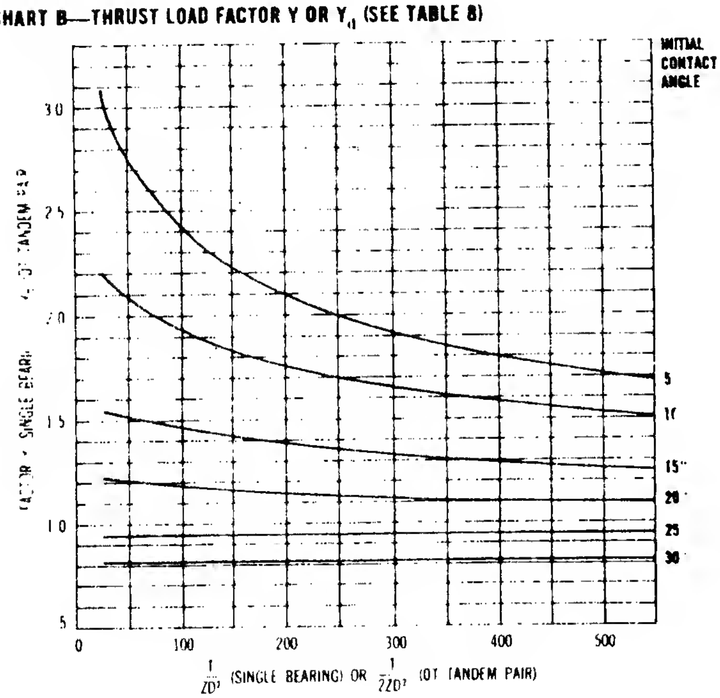
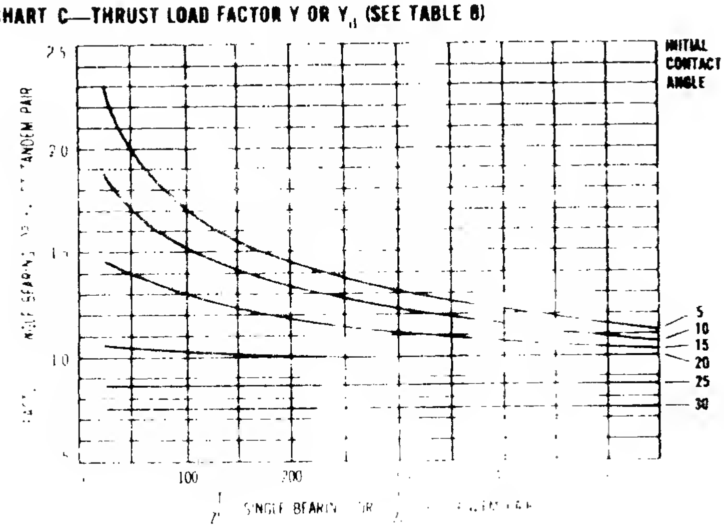


CHART C—THRUST LOAD FACTOR Y OR Y_d (SEE TABLE 8)



Entering Table 8 with 38H, select Chart C as shown. Entering Chart with 105 for $\frac{T}{ZD^2}$ and 17° for contact angle, factor Y = 1.2

Using Formula 2:

$$P = X(R_h + 1.2R_a) + YT$$
$$= .43(4.3 + 2.8) + 1.2(23)$$
$$= 3.1 + 28 = 31.1 \text{ pounds}$$

Formula 2 results in a greater load than Formula 1; in the computation below P = 31.1 pounds, the higher value.

FATIGUE LIFE COMPUTATIONS—HOURS

Since fatigue life is usually computed in terms of hours at a given speed, the dynamic load ratings C_s are given in the dynamic load charts on the listing pages of this catalog for speeds of 100 rpm and higher. The design life basis for C_s values is 500 hours for 90% survival, as developed from load rating evaluation methods standardized by the Anti-Friction Bearing Manufacturers Association.

Formula 3 gives the simplified computation for fatigue life in hours using equivalent radial load value P, as derived above, and dynamic load rating C_s from the listing pages to find the life ratio.

FORMULA 3 (simplified computation)

$$L_{rs} = \frac{C_s}{P}$$

where:
L_{rs} = life ratio
C_s = dynamic radial load rating, pounds
P = equivalent radial load, pounds

Dynamic load rating C_s is found in the dynamic load chart on the appropriate listing page in the front of this catalog. This chart is entered with the speed of rotation and the DATA REFERENCE NUMBER for the bearing selected.

Equivalent radial load P is the larger of the two values resulting from solution of Formulas 1 and 2 (page 61).

Life ratio L_{rs} is used to enter Chart 6 to determine design life or average life.

EXAMPLE—SIMPLIFIED LIFE COMPUTATION

Entering the dynamic load rating chart with 80,000 rpm and DATA REFERENCE NUMBER 38H, C_s = 43 pounds.

This chart shows that the 38H bearing has a speed limit below 80,000 rpm for grease lubrication; it should not be used in this application without oil spray or mist.

Using Formula 3: $L_{rs} = \frac{C_s}{P} = \frac{43}{31.1} = 1.38$

Entering Chart 6 with L_{rs} of 1.38 design life L_{10} is approximately 13[illegible] hours and average life is approximately 6500 hours.

Formula 4 gives the full computation for fatigue life in hours, using value P and ratings C to find life in hours directly. This will give ap

roximately the same value derived from Formula 3, but with greater accuracy.

FORMULA 4 (full computation)

$$L_{10} = 500\left(\frac{C_a}{P}\right)^3$$

where: L_{10} = life in hours

C_a = dynamic radial load rating

P = equivalent radial load, pounds

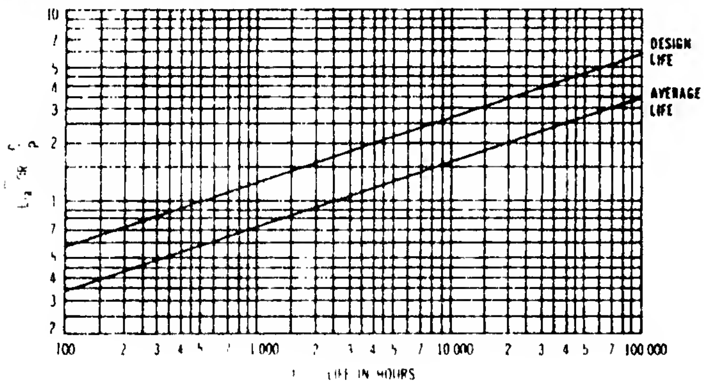
Factors C_a and P are identical to those used in Formula 3.

EXAMPLE—FULL LIFE COMPUTATION

$$L_{10} = 500\left(\frac{C_a}{P}\right)^3 = 500\left(\frac{43}{31.1}\right)^3$$

$$= 500(1.38)^3 = 1314 \text{ hours}$$

CHART 6—DESIGN OR AVERAGE FATIGUE LIFE—HOURS



FATIGUE LIFE COMPUTATION—REVOLUTIONS

Life computation in revolutions uses basic dynamic radial load ratings C, shown in Table 8, which are based on a life of 1,000,000 revolutions with 90% survival being the equivalent of 500 hours design life at 33-1/3 rpm only.

The C rating for a given bearing is greater than the C_o rating, the static radial load rating given on the listing page. It would therefore be unwise to use only rating C for life computation in applications operating at low speeds where shock or vibratory loads may be involved. Another limitation of using C ratings is that they do not give consideration to speed limits in terms of lubrication, ball retainer type and centrifugal ball loading effects on capacity at very high speeds.

Formula 5 may be used when life in revolutions is needed for reference or for purposes of comparison.

FORMULA 5 $\quad L_{10r} = \left(\frac{C}{P}\right)^3 \times 10^6$ revolutions

where: L_{10r} = design life in revolutions

C = basic dynamic radial load rating, pounds

P = equivalent radial load, pounds

DUPLEX BEARING LIFE COMPUTATION

For DT tandem duplex pairs, equivalent radial load ratings are found by solving Formulas 6 and 7. The larger of the two values derived is used in Formula 3 or 4 to find the design fatigue life of the bearing pair selected.

FORMULA 6 $\quad P = .62\,(R_h + 1.2R_s)$

FORMULA 7 $\quad P = .62X\,(R_h + 1.2R_s) + .62Y_dT$

where: P = equivalent radial load of tandem pair, pounds

R_h = radial load fixed in relation to outer rings, pounds

R_s = radial load fixed in relation to inner rings, pounds

X = radial load factor

Y_d = thrust load factor of tandem pair

T = thrust load, pounds

The factor .62 takes account of slight errors in load division between the two bearings and the probability that one bearing of the pair may fail before the other, making advisable the replacement of both bearings of the pair for maximum reliability in later operation.

Factors R_h, R_s and X are identical to those used in Formulas 1 and 2 for individually mounted bearings.

To obtain factor Y_d, first compute value $\frac{T}{2ZD^2}$. ZD^2 is found in Table 8 under the DATA REFERENCE NUMBER of the single bearing to be used in a duplex DT pair. Select Chart B or Chart C as shown by Table 8 and enter with $\frac{T}{2ZD^2}$ and the contact angle of the bearing to determine factor Y_d.

Formulas are not shown for computing life of DB or DF preloaded pairs since preload requirements, rather than bearing capacity, often limit the amount of external loading that may be applied safely. This is particularly true for miniature and instrument bearings, less true for spindle and turbine bearings. However, speed capability of spindle and turbine preloaded pairs is generally limited by energy loss, heating or by type of lubrication. Users of duplex preloaded pairs for new applications are urged to consult Barden for complete analysis and bearing recommendations.

INCREASED LIFE RELIABILITY

If desired, factors for use with dynamic ratings will be supplied by Barden for computation of fatigue life on the basis of reliability greater than 90% survival. However, at high speeds and high temperatures many other factors must be considered, such as accuracy of mounting, degree of dynamic balance, effectiveness of lubrication system and thermal constraining influences. A complete review of requirements by Barden engineers is recommended to attain the increased life reliability desired.

.BLE 8—VALUES FOR FATIGUE LIFE COMPUTATION

MINIATURE AND INSTRUMENT BEARINGS

DATA REFERENCE NUMBER	Chart used for factor Y (page 62)	INITIAL CONTACT ANGLE—degrees						Ball complement		Value ZD^2	Basic dynamic load rating C pounds
		Radial play range						Number Z	Diameter D		
		Code 2	Code 3	Code 4	Code 5	Code 6	Std.				
R0	B	12	17	18	25	27		6	1/32"	.0059	21
R1	B	11	14	16	22	26		6	1 mm	.0093	30
R1-5B	B						16	6	1/16"	.0234	60
R2	B	9	11	12	16	20		7	1/16"	.0273	69
R2B	B						16	7	1/16"	.0273	69
R2H	C						16	8	1/16"	.0312	92
R2-5	B	9	11	12	16	20		6	1/16"	.0234	60
R2-5B	B						20	6	1/16"	.0234	60
R2-5H	B						12	7	1/16"	.0273	66
R2-6	B	9	11	12	16	20		7	1/16"	.0273	80
R3	B	7	9	10	13	16		7	3/32"	.0615	140
R3B	B						16	7	3/32"	.0615	140
R3H	B						10	8	3/32"	.0703	153
R3W	B	7	9	10	13	16		6	3/32"	.0528	127
R4	B	7	9	10	13	16		8	3/32"	.0703	159
R4B	B						16	8	3/32"	.0703	159
R4H	B						10	9	3/32"	.0791	172
R8	C	12	15	17	22	27		10	5/32"	.244	748
R133	B	12	17	18	25	27		7	1/32"	.0068	23
R133W	B	12	17	18	25	27		8	1/32"	.0078	26
R144	B	11	14	16	22	26		8	1 mm	.0124	39
R156	B	11	14	16	22	26		9	1 mm	.0140	40
R156W	B	11	14	16	22	26		11	1 mm	.0171	46
R166	B	9	11	12	16	20		8	1/16"	.0312	89
R168	B	11	14	16	22	26		11	1 mm	.0171	40
R188	B	9	11	12	16	20		11	1/16"	.0430	108
Z114	B		9		13	16		12	3/32"	.105	204
34	B	6	7	9	10	13		6	1/8"	.0938	202
34BX4	B						12	6	1/8"	.0938	164
34H	C			13				8	1/8"	.125	336
34 5B	B						15	6	1/8"	.0938	202
36	B	6	7	8	10	13		6	9/64"	.119	258
36BX1	B						12	6	9/64"	.119	209
36H	C				14	17		8	9/64"	.159	42[illegible]
38	B	6	7	8	10	13		7	5/32"	.171	357
38BX2	B						14	7	5/32"	.171	355
38H	C				14	17		9	5/32"	.220	570
103	C	11	12	16	20	24		10	3/16"	.352	1040

LARGE BORE, EXTRA THIN BEARINGS

BASIC BEARING NUMBER	Chart used for factor Y (page 62)	INITIAL CONTACT ANGLE—degrees		Ball complement		Value ZD^2	Basic dynamic load rating C pounds
		Radial play range		Number Z	Diameter D		
		Code 3	Code 5				
A538T	C	15	22	10	1/8"	.156	495
A539T	C	15	22	12	1/8"	.188	548
A540T	C	15	22	14	1/8"	.219	596
A541T	C	15	22	16	1/8"	.250	629
A542T	C	15	22	18	1/8"	.281	646
A543T	C	15	22	22	1/8"	.344	721

ABLE 8—VALUES FOR FATIGUE LIFE COMPUTATION (Continued)

SPINDLE AND TURBINE BEARINGS

DATA REFERENCE NUMBER	Chart used for factor Y (page 62)	INITIAL CONTACT ANGLE—degrees Radial play range Code 3	Code 4	Code 5	Code 6	Ball complement Number Z	Diameter D	Value ZD^2	Basic dynamic load rating C pounds
34	B	7		10	13	6	1/8"	.0938	202
34H	C		13			8	1/8"	.125	336
36	B	7		10	13	6	9/64"	.119	257
36H	C			14	17	8	9/64"	.159	424
36J	C			14	17	8	9/64"	.159	424
38	B	7		10	13	7	5/32"	.171	357
38H	C			14	17	9	5/32"	.220	570
38J	C			14	17	9	5/32"	.220	570
38X2	C	14		20	25	7	5/32"	.171	570
100	C	12		20	24	7	3/16"	.246	793
100H	C			13	16	9	3/16"	.316	795
100J	C			13	16	10	3/16"	.352	851
100X1	C	10		13	16	7	3/16"	.246	734
101	C	12		20	24	8	3/16"	.281	884
101H	C			13	16	10	3/16"	.352	868
101J	C			13	16	11	3/16"	.386	926
102	C	12		20	24	9	3/16"	.316	970
102H	C			13	16	11	3/16"	.386	936
102J	C			13	16	12	3/16"	.422	990
103	C	12		20	24	10	3/16"	.352	1040
103H	C			13	16	13	3/16"	.457	1050
103J	C			13	16	13	3/16"	.457	1050
104	C	9		13	16	10	1/4"	.625	1620
104H	C			12	15	11	1/4"	.688	1570
104J	C			12	15	13	1/4"	.813	1750
105	C	9		13	16	10	1/4"	.625	1850
105H	C			12	15	13	1/4"	.813	1750
105J	C			12	15	14	1/4"	.875	1830
106	C	9		13	16	11	9/32"	.870	2280
106H	C			12	15	14	9/32"	1.11	2260
106J	C			12	15	14	9/32"	1.11	2260
107	C	8		12	15	11	5/16"	1.07	2750
107H	C			11	14	15	5/16"	1.46	2850
107J	C			11	14	15	5/16"	1.46	2850
200	C	11		16	21	7	7/32"	.335	1040
200H	C			13	16	9	7/32"	.431	1230
200J	C			13	16	9	7/32"	.431	1230
201	C	11		16	21	7	15/64"	.385	1180
201H	C			13	16	9	15/64"	495	1400
201J	C			13	16	10	15/64"	.549	1500
202	C	10		15	20	7	1/4"	.438	1340
202H	C			13	16	10	1/4"	.625	1700
202J	C			13	16	10	1/4"	.625	1700
203	C	10		15	20	8	17/64"	565	1650
203H	C			12	15	10	17/64"	.706	1920
203J	C			12	15	11	17/64"	.776	2050
204	C	10		14	18	8	5/16"	.781	2210
204H	C			12	15	10	5/16"	.976	2570
204J	C			12	15	11	5/16"	1.07	2740
205	C	10		14	18	9	5/16"	.879	2420
205H	C			12	15	11	5/16"	1.07	2760
205J	C			12	15	13	5/16"	1.27	3090
206	C	9		14	18	9	3/8"	1.26	3360
206H	C			12	15	12	3/8"	1.69	4070
206J	C			12	15	12	3/8"	1.69	4070
207	C	8		13	16	9	7/16"	1.72	4420
207H	C			11	14	12	7/16"	2.29	5350
207J	C			11	14	13	7/16"	2.49	5650
208	C	8		13	16	9	15/32"	1.98	[illegible]
208H	C			11	14	12	15/32"	2.64	[illegible]
209	C	8		13	16	10	15/32"	2.20	5370
210	C			11	14	13	15/32"	2.86	6400

LUBRICANTS AND LUBRICATION

oper and adequate lubrication of precision ball bearings is essential if maximum performance is to be obtained. Of equal importance is the exclusion of foreign matter by suitable design of enclosure and, where possible, the use of shielded bearings, even for the most critical low torque applications.

Choice of the proper lubricant is influenced not only by the nature of the application and available means for applying the lubricant, but by the bearing itself.

Lubricants normally supplied are shown on the listing pages. General guides for oil or grease lubrication are presented in Tables 9 and 10. These tables list in summary form the oils and greases most frequently specified, along with their temperature and speed ranges. Other lubricants, some of which are being used successfully under more exacting conditions, can be supplied on request. Consult Barden for assistance in bearing choice, specifications and lubrication recommendations for high speed and high temperature applications.

OIL

Where speeds and loads are very low, such as gyro gimbal, synchro, servomechanisms and computer gear train applications, synthetic instrument oil (MIL-L-6085A, Barden code 0-11) is recommended. For extremely low values of starting or running torque, use of such an oil is essential. factory prelubricated, a few milligrams per bearing and up, depending on bearing size, will suffice for many thousands of hours of operation.

With oil lubrication, performance life is mainly dependent on temperature and speed since higher temperatures increase the rate of evaporation and oxidation, and higher speeds tend to dissipate the oil. Extreme caution should be observed in specifying minimal amounts of oil because of shortened life resulting from lubricant starvation; consult Barden when requirements call for less than normal amounts of oil prelubrication.

GREASE

Moderate speed and heavier load applications operate successfully with grease, which is more readily retained than oil within the bearing or housing enclosure. Grease tends to remain in close proximity to the critical load carrying areas and provides lubricating action by bleed-out of oil in the grease.

For a wide variety of motor, generator and control applications, operating at speeds in hundreds and thousands of rpm, bearings prelubricated with grease may be used with complete success. Performance life in such cases usually is limited by oxidation and thermal stability of greases and evaporation of oil content.

Air passage through bearings has an adverse effect on greases and bearing life and therefore should be minimized by all possible means. 'en Flexeal sealed bearings will prolong life in such cases.

Bearings are factory lubricated with grease by filling 20 to 35% of available internal space. Standard quantities meet the needs of most applications. Reduced quantities for unusual requirements can be supplied on special order.

Grease lubricated machine tool spindle applications are successful to about 600,000 dn—product of bearing bore in mm (d) and rpm of spindle (n)—with bearing temperature in the range of 80° to 130°F (25° to 55°C). Since grease is a poor conductor of heat, applications with high operating temperatures or speeds above 600,000 dn usually require oil mist or spray lubrication to achieve reliable bearing life.

VACUUM IMPREGNATED RETAINERS

Bearings for certain classes of inertial gyros, where oil or grease mass shifts may adversely affect the accuracy of the gyro system, may be obtained with ball retainers vacuum impregnated with oil and the bearings centrifuged to remove the excess. Since many variables are involved in the successful use of this method for specific gyro rotor designs, Barden engineers should be consulted for recommendations.

Excellent results are attained in certain applications by vacuum impregnation of phenolic ball retainers with oil and subsequent addition of grease. All oils and greases are not compatible, but in general the oil component of the grease should be the same type as the oil used for impregnation. Consult Barden for advice on best combinations.

OIL SPRAY OR MIST

Advantages of pressurized air-oil mist equipment for machine tool spindle applications include once-through passage of clean oil, superior cooling of critical bearing surfaces and exclusion of foreign substances. oil mist should be directed from the interior of the spindle housing and exhausted outward through the housing seals.

Pressurized air-oil mist systems require a source of clean compressed air, thus are usually too cumbersome for airborne or space vehicles. Spray oil or wick fed systems may be used for many airborne applications but specially designed prelubricated or self-lubricating bearings may be required for space vehicles.

Many simple turbine devices operate successfully with wick feed or induced oil mist lubricating systems where loads are moderate and speed is high. For larger turbine units spray oil systems are preferred to provide adequate cooling and positive lubrication at all times. The oil supply should be well filtered with the cleanest oil directed first through the bearings that operate at highest speeds.

Designers are urged to consult Barden at an early stage for advice and recommendations on high speed lubrication problems.

'AFT AND HOUSING SHOULDERS

, assure best performance of precision ball bearings, shaft and housing shoulders must be high enough to provide accurate and solid seating, good alignment and support under maximum thrust load conditions. At the same time, shoulders should avoid possible interference with ball retainers, shields or seals when bearings have maximum values of radial play and are under heavy thrust loads.

If the design permits, shoulders should be low enough to allow bearing tools tc be used against appropriate ring faces when bearings are dismounted, thus avoiding damage to raceways from forces transmitted through ball contacts. This is particularly necessary for bearings that are interference fitted, if further service from the bearings is needed subsequent to dismounting.

Spacers, sleeves or other parts may also be used to provide shoulders as long as recommended dimensional limits are observed. If thrust loads are small, snap rings or circlips may be used. However, it is preferable that the rotating ring of the bearing be located against an accurately machined surface on at least one face.

Shaft shoulder diameters should not be less than S_{min} and not exceed L_i for open bearings or U_i for shielded or sealed bearings. Housing shoulder diameters should not exceed H_{max} and not be less than L_o for open bearings or U_o for shielded or sealed bearings. See Figure 14, below. Dimensions L_i, L_o, U_i, and U_o are found on the listing pages for bearing selected.

Under certain conditions larger shoulder heights are permissible; where required because of design problems consult Barden for recommendations.

On high speed applications where oil spray or mist systems are employed, shoulder design may be extremely important as it is essential that lubricant flow be unimpeded and effective. Consultatio with Barden at an early design stage is invited.

Table 11 gives minimum shaft and maximum housing shoulder diameters S_{min} and H_{max} arranged according to bearing bore and O.D., and maximum shaft or housing fillet radius dimensions found on the listing pages.

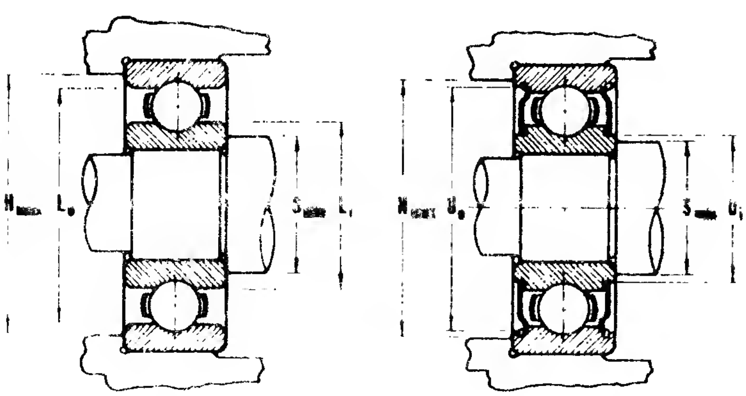


FIGURE 14—Shaft and housing diameters for open bearings (left) and shielded or sealed bearings (right).

TABLE 11—SHAFT AND HOUSING SHOULDERS

MINIATURE AND INSTRUMENT BEARINGS

BEARING DIMENSIONS Inches		MAXIMUM FILLET RADIUS Inches	SHAFT AND HOUSING DIMENSIONS Inches	
Bearing bore d	Bearing O.D. D	r	Minimum shaft shoulder S_{min}	Maximum housing shoulder H_{max}
.0469	.1562	.005	.070	.132
.0550	.1875	.005	.080	.162
.0781	.1875	.005	.110	.166
.0781	.2500	.005	.110	.220
.0937	.1875	.005	.115	.166
.0937	.2500	.005	.120	.220
.0937	.2883	.005	.115	.258
.0937	.3125	.005	.124	.282
.1250	.2500	.005	.153	.220
.1250	.2883	.005	.153	.258
.1250	.3125	.005	.153	.282
.1250	.3750	.005	.153	.341
.1250	.3750	.012	.170	.325
.1250	.4100	.005	.153	.370
.1250	.4250	.005	.153	.385
.1250	.4375	.005	.153	.400
.1250	.5000	.005	.153	.460
.1250	.5000	.012	.170	.440
.1250	.5769	.005	.153	.525
.1250	.7500	.005	.153	.690
.1562	.3125	.005	.187	.284
.1575	.6299	.012	.220	.570
.1875	.3125	.005	.215	.284
.1875	.3750	.005	.215	.341
.1875	.5000	.005	.215	.450
.1875	.5000	.012	.240	.450
.1875	.5694	.012	.240	.520
.1875	.7435	.012	.240	.690
.1875	.7500	.012	.240	.700
.1875	.7717	.012	.240	.710
.1875	.8685	.012	.240	.805
.1875	.8750	.012	.240	.810
.1969	.6299	.012	.250	.570
.1969	.7480	.012	.260	.680
.2362	.7480	.012	.300	.680
.2500	.3750	.005	.276	347
.2500	.5000	.005	285	.450
.2500	.6250	.012	.310	.560
.2500	.7500	.016	.320	.670
.2500	1.0480	.012	.310	.970
.2756	.8661	.012	.350	790
.3150	8661	.012	390	790
.3750	.8750	.016	.450	.800
.5000	8750	.016	.580	800
.5000	1.1250	.016	.580	1.050
.6250	1.3750	.031	.750	1.250

SHAFT AND HOUSING FITTING PRACTICE

Improper fitting of precision bearings is almost certain to result in poor performance. If the bearing seat is out of round or if the fit is too tight, dimensional accuracy of the raceway may be seriously disturbed, or the carefully controlled internal clearance in the bearing may be destroyed. Too loose a fit, on the other hand, may prevent proper functioning because of lack of control of the rotating component.

Interference fits should be approached with caution, particularly with thin-section bearings, as they can cause loss of internal clearance or force the raceway out of round with resulting noisy or poor operation. Close clearance fits are usually recommended to avoid distortion or damage to bearings, but they result in looseness that may be objectionable for certain applications. This effect can be eliminated by suitable face clamping or locking means.

Interference fits are seldom necessary on miniature and instrument bearings except where loads are appreciable or when unusually close control of mass shift is required. On spindle and turbine bearings there is more frequent need for interference fits because of heavier radial loading and hot-running conditions at high speed.

While an interference fit will prevent fretting corrosion and wear of the seating surface through relative movement, only one ring of a bearing should normally be interference fitted. The other ring should be free to move axially on the housing or shaft seat to relieve thermal expansion or contraction forces or to permit proper axial adjustment or spring preloading.

Problems are sometimes encountered with outer rings turning in stationary housings. This is usually due to rotating radial loads created by dynamic unbalance. Better balancing will usually correct such a condition, providing a sounder solution than interference fitting of the outer ring in the housing.

Preloaded bearing pairs require clearance fits for outer rings in stationary housings to avoid substantial increase in preload from internal heat generation or to permit the non-fixed or "floating" pair to move axially for relief from thermal constraint.

High speed turbine applications also may require more clearance in the housing, particularly where some heat is transmitted from the shaft through the bearing in addition to heat generated within the bearing due to the high speed.

Although soft non-ferrous housings may be used for lightly loaded applications at normal temperatures, steel housing liners should be used for all applications where there is a wide range of operating temperatures. A steel liner will resist contraction of the housing on the outer ring of the bearing at low temperatures and minimize excessive loosening of the fit at high temperatures.

Examples of shaft and housing fits

Examples of typical shaft and housing fits, based on working tolerance ranges on both bearings and mounting parts, are shown in Table 12. The extremes of resulting fits, while possible in rare instances, are seldom encountered in actual practice. However, Table 12 shows how such extremes of fits can be reduced by selective assembly of calibrated bearings and mounting members, which is described in the Calibration section that follows.

Because of the wide variety of applications, as well as loads, speed and environmental conditions, Table 12 provides only a generalized guide to mounting fits. Complete recommendations as to proper fits are available from Barden for applications where previous experience does not indicate clearly the type of fit and mounting arrangement that should be used.

Computations are given on pages 72 and 73 for determination of shaft and housing size limits for a given bearing size and tolerance range, using either calibrated or non-calibrated bearings.

ABLE 11—SHAFT AND HOUSING SHOULDERS

SPINDLE AND TURBINE BEARINGS

BEARING DIMENSIONS Inches		MAXIMUM FILLET RADIUS Inches	SHAFT AND HOUSING DIMENSIONS Inches	
Bearing bore d	Bearing O.D. D	r	Minimum shaft shoulder S_{min}	Maximum housing shoulder H_{max}
.1575	.6299	.012	.220	.570
.1969	.6299	.012	.250	.570
.1969	.7480	.012	.260	.680
2362	.7480	.012	.300	.680
.2756	.8661	.012	.350	.790
.3150	.8661	.012	.390	.790
.3150	.9449	.016	.390	.870
.3543	1.0236	.016	.450	.930
.3937	1.0236	.012	.470	.950
3937	1.1811	.025	.500	1.050
4724	1.1024	.012	.550	1.020
4724	1.2598	.025	.580	1.130
5118	1.2598	.025	.620	1.130
.5906	1.2598	.012	.670	1.180
.5906	1.3780	.025	.710	1.250
6693	1.3780	.012	.750	1.300
6693	1.5748	.025	.790	1.450
7874	1.6535	.025	.910	1.520
.7874	1.8504	.040	.960	1.670
.9843	1.8504	.025	1.100	1 720
9843	2.0472	.040	1.160	1.860
1.1811	2.1654	.040	1.350	1.990
1.1811	2.4409	.040	1.360	2.240
1.3780	2.4409	.040	1.550	2.240
1.3780	2 8346	.040	1 570	2.600
1.5748	3.1496	.040	1 770	2.910
1 7717	3 3465	040	1.980	3.100

LARGE BORE, EXTRA THIN BEARINGS

6250	1 0625	015	.710	970
7500	1 1875	015	840	1.100
.8750	1.3125	.015	.960	1.220
1.0625	1.5000	015	1.150	1.410
1 3125	1.7500	.015	1.400	1.660
1 5625	2.0000	015	1.650	1.910

Flanged bearing mounting

Use of flanged miniature and instrument bearings has many advant[illegible] where the design of the application will permit this method of mount ing. As shown by Figure 15, flanged bearings provide solid mounting for axial control and obviate the need for housing shoulders or shoulder rings. Their use permits through-boring of housings to reduce manufacturing cost and simplify assembly. Mountings for flanged bearings require accurately machined faces on housings to position and support the bearings.

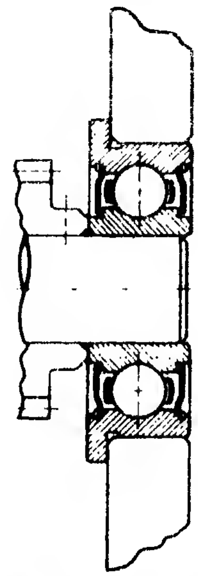


FIGURE 15—Flanged bearings permit use of through-bore housings, eliminating need for shoulders inside housing bore, and provide solid mounting for axial control.

Maximum fillet radii

When integral shaft or housing shoulders are used, fillet radii of shoulders must be made to clear corner radii of bearing inner ring and outer ring, respectively, to allow accurate seating of bearings against shoulders. Value r given in Table 11 and on the bearing listing pages is the maximum shaft or housing fillet radius that bearing corners will clear. On angular contact bearings the smaller dimensions r_i or r_o instead of r, should be used as the maximum shaft or housing fillet radius when non-thrust faces of outer or inner rings are mounted against solid shoulders.

Figure 16 shows how observance of this dimension provides the necessary clearance. Also shown in Figure 16 is the use of an undercut fillet which is preferred wherever possible since it facilitates accurate machining of the shoulder and bearing seat, and permits accurate seating of the bearing against the shoulder.

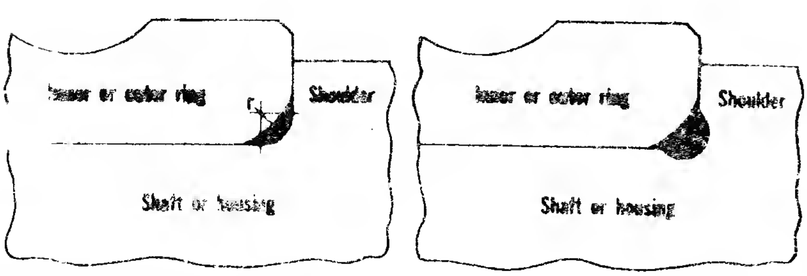


FIGURE 16—Two methods of providing clearance for bearing corner. [illegible] radius r is maximum that bearing corner will clear. (right) undercut fillet allows more accurate machining of shoulder and seat, provides more accurate bearing mounting.

TABLE 12—SHAFT AND HOUSING FITS

MINIATURE AND INSTRUMENT BEARINGS

		Dominant requirements	FIT EXTREMES—inches* Random fitting	Selective fitting	Typical applications
Shaft fits	Inner ring clamped	Normal accuracy	.0000 .0004 (loose)	-.0001 (loose) -.0003 (loose)	Tape guide rollers, textile spindles
		Very low runout, high radial rigidity	+.0001 (tight) -.0003 (loose)	.0000 —.0002 (loose)	Gyro rotors
	Inner ring not clamped	Normal accuracy	+.0001 (tight) —.0003 (loose)	.0000 —.0002 (loose)	Synchros, small motors, servos, potentiometers, gyro gimbals, computers
		Very low runout, high radial rigidity	+.0003 (tight) —.0001 (loose)	+.0002 (tight) .0000	Gyro rotors
		Very high speed service	+.0002 (tight) .0002 (loose)	+.0001 (tight) -.0001 (loose)	Blower motors, textile spindles
		Inner ring must float to allow for expansion	.0000 .0004 (loose)	.0001 (loose) .0003 (loose)	Clutches, brakes
		Inner ring must hold fast to rotating shaft	+.0003 (tight) .0001 (loose)	+.0002 (tight) .0000	Motors, generators, gear drives, dental drills
Housing fits		Normal accuracy, low to high speeds Outer ring can move readily in housing for expansion	.0000 -.0004 (loose)	.0001 (loose) .0003 (loose)	Motors, servos, generators, blowers, dental drills, tape spindles, potentiometers
		Very low runout, high radial rigidity Outer ring need not move readily to allow for expansion	+.0001 (tight) .0003 (loose)	.0000 —.0002 (loose)	Gyro gimbals, synchros, resolvers, platform gimbals, computers
		Heavy radial load Outer ring rotates	+.0001 (tight) —.0003 (loose)	.0000 —.0002 (loose)	Rollers, pivots
		Outer ring must hold fast to rotating housing Outer ring not clamped	+.0004 (tight) .0000	+.0003 (tight) +.0001 (tight)	Cam rollers, pulleys, idler gears

*Tight fits are positive (+) and loose fits negative (—) for use in shaft and housing size determination, pages 72 and 73

SPINDLE AND TURBINE BEARINGS

		Dominant requirements		FIT EXTREMES—inches*	Typical applications
Shaft fits	Inner ring clamped	Very low runout, high radial rigidity		+.00005 (tight) .00025 (loose)	Grinding spindles, magnetic drums, platform gimbals
		Low to high speeds, low to moderate radial loads		+.00005 (tight) .00025 (loose)	Turbines, compressors, motors, generators
		Heavy radial load	Inner ring rotates	+.00005 (tight) +.00035 (tight)	Gear or belt loaded applications
			Outer ring rotates	.0000 .0003 (loose)	Idler or planet gears
	Inner ring not clamped	Very low runout, high radial rigidity		+.00005 (tight) +.00035 (tight)	Magnetic drums, grinding spindles, platform gimbals
		Moderate to high speeds, light to moderate radial loads		+.00005 (tight) +.00035 (tight)	Turbines, compressors, motors, generators, gear shafts
		Heavy radial load, low to moderate speeds	Inner ring rotates	+.00005 (tight) +.00035 (tight)	Gear or belt loaded applications
			Outer ring rotates	.0000 —.0003 (loose)	Idler or planet gears
		Inner ring must float to allow for expansion low speed only		.00005 (loose) .00035 (loose)	Platform gimbals
Housing fits		Normal accuracy, low to high speeds, moderate temperature		.0000 —.0004 (loose)	Spindles, magnetic drums, motors, generators
		Very low runout, high radial rigidity Outer ring need not move readily to allow for expansion		+.0001 (tight) .0003 (loose)	Platform gimbals, resolvers, inductosyns
		High temperature, moderate to high speed Outer ring can move readily to allow for expansion		.0001 (loose) .0005 (loose)	Turbines, compressors, starters
		Heavy radial load Outer ring rotates		+.0002 (tight) .0002 (loose)	Idler or planet gears, cam rollers

*Tight fits are positive (+) and loose fits negative (—) for use in shaft and housing size determination, pages 72 and 73

librated bearings, also known as classified or graded bearings, make possible extremely close control of shaft and housing fits through use of selective assembly methods in installing bearings. With bearing bores and mating shaft diameters segregated into two or more groups within the normal tolerance range, and with bearing O.D.s and mating housing diameters similarly calibrated, the tolerance buildup on shaft and housing fits can be materially reduced. This achieves the same result as reducing tolerances of bearings and mating parts, but at far less cost.

Table 12 (page71) shows the range of fits obtained from miniature and instrument bearings and mating parts chosen at random for non-selective assembly as compared with use of calibrated bearings and mating parts in selective assembly. Computations for determining shaft and housing size limits for calibrated and non-calibrated bearings are given below.

Table 13 lists the standard Barden Precision calibration groups which apply to all standard bearings shown in this catalog except the A500 large bore, extra thin sizes. Table 14 shows standard calibration groups for Barden A500 series bearings only. Each bearing is calibrated on the basis of minimum bore and maximum O.D. size to correlate with maximum shaft and minimum housing size where out-of-roundness exists.

Standard two-group calibration of both bores and O.D.s is specified by the suffix C. For identification in users' plants, such calibration groups shown on Barden bearing packages as:

C11 = Bore 1, O.D. 1	C21 = Bore 2, O.D. 1
C12 = Bore 1, O.D. 2	C22 = Bore 2, O.D. 2

Calibration on bores only is ordered by suffix CX0. Calibration on O.D.s only is ordered by suffix C0X. On package markings the appropriate group number 1 or 2 is substituted for the X ordering symbol, as C10 or C01, etc.

TABLE 13—BARDEN PRECISION CALIBRATION CODES

BORE CALIBRATION

When tolerance is	
+.0000" to .0002"	+.0000" to .0001" = Bore 1
	−.0001" to −.0002" = Bore 2

When tolerance is	
+.0000" to −.00015"	+.0000" to −.0001" = Bore 1
	−.0001" to −.00015" = Bore 2

O.D. CALIBRATION

When tolerance is	
+.0000" to −.0002"	−.0000" to −.0001" = O.D. 1
	−.0001" to −.0002" O.D. 2

When tolerance is	
+.0000" to .0003"	+.0000" to .0001" O.D. 1
	.0001" to .0002" O.D. 2
	.0002" to .0003" O.D. 3

TABLE 14—BARDEN A500 CALIBRATION CODES

BORE CALIBRATION

When tolerance is	
+.0000" to −.0003"	+.0000" to −.00015" = Bore 1
	−.00015" to −.0003" = Bore 2

O.D. CALIBRATION

When tolerance is	
+.0000" to −.0004"	+.0000" to −.0002" = O.D. 1
	−.0002" to −.0004" = O.D. 2

SHAFT SIZE DETERMINATION

Manufacturing dimensions for shafts can be determined from the following computation:

Maximum shaft size = (minimum bearing bore) + (tightest fit extreme)

Minimum shaft size = (maximum bearing bore) + (loosest fit extreme)

Maximum and minimum bearing bores are found by taking the nominal bore size shown on the appropriate listing page of this catalog and adding (algebraically) the extremes of bore tolerance for that bearing as given in the appropriate table in the Tolerances section. For selective fitting, the extremes of bore tolerance to be added are shown in Table 13 or 14 above.

Tightest and loosest shaft fit extremes are found in Table 12 (page tight fits are positive (+), loose fits are negative (—).

EXAMPLE—SHAFT SIZE DETERMINATION

Having selected the SR3SS bearing for a gyro gimbal, the nominal bore size shown on the listing page for this bearing is .1875".

The Barden tolerance range for this bore size shown in Table 1 (ABEC-7) is +.0000" to .0002" for random fitting; for selective fitting, as shown in Table 13, Bore 1 range is +.0000" to .0001" and Bore 2 range is .0001" to .0002".

Fit extremes for a gyro gimbal application in Table 12 are +.0001" (tight) and .0003" (loose) for random fitting, or .0000" and −.0002" (loose) for selective fitting.

Dimension	Random fitting	Selective fitting Group 1	Group 2
Nominal bearing bore	.1875"	.1875"	.1875"
Add: low extreme of bore tolerance	−.0002"	.0001"	−.0002"
Minimum bearing bore	.1873"	.1874"	.1873"
Add: tightest fit extreme	+.0001"	.0000"	.0000"
MAXIMUM SHAFT SIZE	.1874"	.1874"	.1873"
Nominal bearing bore	.1875"	.1875"	.1875"
Add: high extreme of bore tolerance	.0000"	.0000"	−.0001"
Maximum bearing bore	.1875"	.1875"	.1874"
Add: loosest fit extreme	.0003"	.0002"	.0002"
MINIMUM SHAFT SIZE	.1872"	.1873"	.1872"

HOUSING BORE DETERMINATION

Manufacturing dimensions for housing bores can be determined in similar fashion from the following computation:

Minimum housing bore = (maximum bearing O.D.) - (tightest fit extreme)

Maximum housing bore = (minimum bearing O.D.) — (loosest fit extreme)

Maximum and minimum bearing O.D.s are found by taking the nominal O.D. size shown on the appropriate listing page of this catalog and subtracting (algebraically) the extremes of O.D. tolerance for that bearing as given in the appropriate table in the Tolerances section. For selective fitting, the extremes of O.D. tolerance to be added are shown in Table 13 or 14.

Tightest and loosest housing fit extremes are found in Table 12 (page 71); as in the case of shaft fits, tight fits are positive (+), loose fits are negative ( ).

EXAMPLE—HOUSING BORE DETERMINATION

The nominal O.D. size shown on the listing page for bearing SR3SS is .5000″.

The Barden tolerance range for this O.D. size shown in Table 1 (ABEC-7) is +.0000″ to .0002″ for random fitting; for selective fitting, as shown in Table 13, O.D. 1 range is +.0000″ to -.0001″ and O.D. 2 range is .0001″ to .0002″.

Fit extremes for the gyro gimbal application in Table 12 are +.0001″ (tight) and .0003″ (loose) for random fitting, or +.0000″ (tight) and .0002″ (loose) for selective fitting.

Dimension	Random fitting	Selective fitting Group 1	Selective fitting Group 2
Nominal bearing O.D.	.5000″	.5000″	.5000″
Add: high extreme of O.D. tolerance	.0000″	.0000″	--.0001″
Maximum bearing O.D.	.5000″	.5000″	.4999″
Subtract: tightest fit extreme	+.0001″	.0000″	.0000″
MINIMUM HOUSING BORE	.4999″	.5000″	.4999″
Nominal bearing O.D.	.5000″	.5000″	.5000″
Add: low extreme of O.D. tolerance	.0002″	-.0001″	—.0002″
Minimum bearing O.D.	.4998″	.4999″	.4998″
Subtract: loosest fit extreme	- .0003″	.0002″	-.0002″
MAXIMUM HOUSING BORE	.5001″	.5001″	.5000″

TASK CORPORATION II (B)

Following the failure of the 204SST5 bearing, Mr. Wireman used Barden Company catalog data to calculate the manufacturer's design life of the bearing. According to his calculations, the bearing had a "B-10" life of 6,500 hours. He then sent the failed bearing to Barden for examination. Barden replied with recommendation that a bearing with higher load capacity but identical external dimensions, either number M204BJHX2 or 204HJB1519 be used. Mr. Wireman decided to use the 204HJB1519.

The conclusion of the Barden Corporation upon examination of the 204SST5 bearings from the first motor was that failure in the front bearing had been due to poor lubricity between balls and races. This opinion was expressed in a letter from Barden of September 11, 1963, which is attached as Exhibit 1. The answer, Barden asserted, was to use a bearing of higher load capacity and, therefore, longer design life. Of the two bearings suggested by Barden, Mr. Wireman chose the 204HJB1519 because it was much the cheaper, costing only a few cents more than the previous 204SST5 bearing. The 204HJB1519 had a higher initial contact angle[1] (18° instead of 14°) and more balls (11 balls instead of 8) than the previous front bearing, and a design life of 19,000 hours, according to Mr. Wireman's calculations.[2] A Barden catalog page giving specifications for the HJB1519 appears as Exhibit 2.

Mr. Wireman was puzzled as to why it should be necessary to use a bearing design life so much higher than the required operational life, but somehow the idea of using a heavier bearing seemed reasonable. Experience with another motor also lent support to this approach. The other motor was an air-cooled version which attached in the same way to run the same pump. It differed in that although its front shaft bearing was fully immersed, the oil flow path was around, not through the bearing, and there was a seal which kept the hydraulic fluid out of the case, so the rotor and rear shaft bearing were not immersed. The frame also differed by having cooling fins and by allowing room for a larger front bearing, a Barden 205 double-shielded steel retainer bearing which had, according to Mr. Wireman's calculations, a lower contact angle but a design life of approximately 200,000 hours under the same loading conditions as the liquid-cooled pump motor. Another difference of the air-cooled motor was that it was used with MIL-5606 hydraulic fluid (a specification for which appears in Exhibit 3), instead of Skydrol.

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1. The higher angular contact was produced by increasing the radial clearance in the bearing.

2. The 1519 in the bearing number means diametral allowance between balls was held between 15 and 19 ten thousandths of an inch.

Prepared in the Design Division, Department of Mechanical Engineering, Stanford University, by Karl H. Vesper as a basis for student engineering problems. Assistance from Jack Wireman, Elmer F. Ward, Dino Morelli and Thomas Barish is gratefully acknowledged.

No bearing failures had occurred on endurance tests of 2,500 hours with the air-cooled motor.

Since the use of a heavier duty bearing seemed reasonable, and since the 204HJB1519 could be installed in place of the 240SST5 without otherwise altering the motor (the 205 bearing could not) and without much affecting costs, the decision was made to install it in all the liquid-cooled pump motors for Geyser Pump. Both qualification test motors were fitted with the 204HJB1519 and twenty production motors which had been shipped were called back, torn down and reassembled with the higher capacity bearing, a process which consumed about two man hours per motor. Drawings were modified to require the 204HJB1519 on all subsequent units.

Consequently, there was considerable surprise and dismay when it was found after 700 hours running on the reassembled first qualification unit that the new bearing was already starting to fail. This motor had been put back on test to complete the original 2,500 hours. Since the first bearing had failed at 1,800 hours, there were 700 hours remaining of the test. After running the 700 uneventful hours with the new bearing, the motor was stopped and disassembled to inspect those parts which had now completed 2,500 hours. The front bearing, which had as yet shown no symptoms of difficulty while running, was found to be just slightly spalled, signifying the onset of failure. Pictures of this bearing appear in Exhibit 4. A written description of the failed bearings is given in Exhibit 5. This description, dated December 9, 1963, was prepared by an independent consultant, Mr. Thomas Barish, a noted mechanical engineering consultant and author of over twenty published articles on bearings.

C O P Y

EXHIBIT I

THE BARDEN CORPORATION
Barden Precision Ball Bearings
Danbury, Connecticut

September 11, 1963

Mr. Elmer Ward, Chief Engineer
Task Corporation
1009 East Vermont
Anaheim, California

Dear Mr. Ward:

This will summarize our recent telephone conversations and letters regarding the failure of a Barden Precision 204SST5 bearing in a pump motor qualification unit and our recommendations to prevent recurrence of such failures. The pump runs at 6000 rpm in Skydrol at 150°F with a thrust load of 75 lbs on the 204SST5 bearing and 20 lbs on the opposed bearing, a 203SS5. The failure in the 204 size bearing was experienced at 1800 hours.

Examination of the 203SS5 bearing showed that it operated with normal contact angle in the presence of considerable contamination. There is also a very light running band where the bearing apparently operated under a reverse thrust condition. In spite of the evidence of contamination this bearing could have continued operating for a considerable time.

The 204SST5 bearing is a typical fatigue failure due to poor lubricity of the hydraulic fluid. In this type of failure a very fine surface spalling occurs on the inner ring raceway and erodes the metal until increased axial play causes failure by rubbing of the rotor and housing or increased loads due to the roughened raceway cause normal fatigue failure of the balls and outer ring. In this case both of these apparently happened. There is no evidence of inadequate alignment or poor mounting to aggravate the failure. It is doubtful that the contamination seen in the 203 size bearing had any significant effect. It is, however, possible that the use of a double shielded bearing helped keep debris in the ball path and accelerate the failure.

In order to increase the life of the bearing with Skydrol as lubricant and with the loads stated, it would be necessary to increase the capacity of the bearing. Because the failure at 1800 hours was a progressive type failure it would be necessary to make a substantial increase in capacity in order to prevent the application from being continually marginal in operation. The two alternates are first, a duplex pair of bearings which, as you say, would require considerable machine modification and, therefore, were not quoted; or a single bearing of different design for greater capacity. This greater

COPY

Mr. Elmer Ward
Task Corporation

September 11, 1963

capacity could be obtained by using the Barden Precision M204BJHX2 or the 204HJB1519. Both of these bearings have the same envelope dimension as the 204SST5 originally used. The M204BJHX2 was developed particularly for use with heavy loads and low lubricity fluids. It has M50 tool steel rings and balls as shown on drawing SA-3711 which has been forwarded to you. The open, unshielded construction which permits an added flow of fluid which would help wash away the contamination or wear particles which might otherwise collect on the bearings. The oversize balls, high radial play, and high shoulders give the bearing a capacity slightly higher than that of the 205J bearing shown in the Barden Catalogue. Based on the experience with the 204SST5 bearing there should be no problem meeting the desired life of 6000 hours with this bearing.

If, for economic reasons, the M204BJHX2 bearing cannot be used, the next recommendation is the Barden Precision 204HJB1519 bearing. This bearing, because of its larger ball complement than the 204SST5, open construction, and high radial play, should give significantly longer life than the 204SST5.

The design life of the 204SST5 bearing is 26,500 hours, for the 204HJB1519 it is 98,000 hours, and for the M204BJHX2 it is 235,000 hours. Assuming the failure at 1800 hours to be typical, this would give an expected life of 6600 hours for the 204HJB1519 and 16,000 hours for the M204BJHX2. This figure will be increased or the chances of attaining it will be much better with the use of the open, unshielded 204HJB1519 but it would be impossible to put a quantitative figure on this relationship without a considerable amount of testing. If the filtering system can be improved to effectively filter the fluid to a 10 micron level this should also have a beneficial effect on life.

In this discussion we have recommended use of bronze retainers based on your experience with this material in Skydrol. We have no field data on this and would normally have recommended comparative checks with phenolic and bronze. The bronze retainer has the added advantage of an extra ball in the complement, giving increased design life. The material used in these retainers is continuous cast Asarcon (80% copper, 10% tin, 10% lead) bronze. We have found it to be superior to all other types of bronze for use in high performance precision ball bearings.

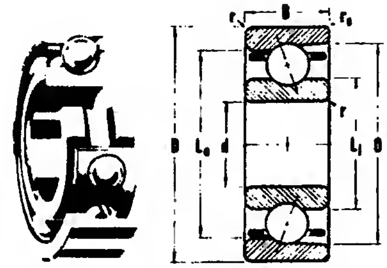
Summing up, the failure of the 204SST5 bearing is a typical low lubricity fluid failure which we feel can be corrected to give good reliability at 6000 hours by use of the Barden Precision 204HJB1519 bearing. It is further suggested that, in the qualification unit, the machine be disassembled at 3000 hours for bearing analysis at Barden, reassembled, and continued for the full life of the bearing, with periodic checks.

BARDEN PRECISION SPINDLE AND TURBINE BEARINGS

Exhibit 2 - Bearing Data

HIGH TEMPERATURE MODERATE TO ULTRA HIGH SPEED

ANGULAR CONTACT BRONZE RETAINER



AVAILABILITY.
All sizes listed are normally available from stock in chrome steel.

■ **Sizes also available in stainless steel; specify by adding prefix S, as S38HJB, etc.**

DIMENSIONS

BORE, O.D., WIDTH d mm	d inches	D mm	D inches	B mm	B inches	MAX. FILLET RADIUS—inches r	r_a	BASIC ORDERING NUMBER	OTHER DIMENSIONS inches L_i	L_o	D	APPROX. WEIGHT pounds	DATA REFERENCE NUMBER
6	.2362	19	.7480	6	.2362	.012	.010	36HJB	.383	.596	.636	.027	36J
7	.2756	22	.8661	7	.2756	.012	.010	37HJB	.463	.692	.739	.035	38J
8	.3150	22	.8661	7	.2756	.012	.010	■ 38HJB	.463	.692	.739	.034	38J
10	.3937	26	1.0236	8	.3150	.012	.010	■ 100HJB	.583	.837	.902	.05	100J
10	.3937	30	1.1811	9	.3543	.025	.015	■ 200HJB	.656	.953	1.028	.08	200J
12	.4724	28	1.1024	8	.3150	.012	.010	101HJB	.670	.924	.989	.06	101J
12	.4724	32	1.2598	10	.3937	.025	.015	201HJB	.721	1.040	1.122	.10	201J
15	.5906	32	1.2598	9	.3543	.012	.010	102HJB	.798	1.053	1.117	.08	102J
15	.5906	35	1.3780	11	.4331	.025	.015	■ 202HJB	.815	1.153	1.240	.12	202J
17	.6693	35	1.3780	10	.3937	.012	.010	■ 103HJB	.895	1.153	1.217	.09	103J
17	.6693	40	1.5748	12	.4724	.025	.015	203HJB	.952	1.292	1.394	.18	203J
20	.7874	42	1.6535	12	.4724	.025	.015	■ 104HJB	1.050	1.390	1.474	.17	104J
20	.7874	47	1.8504	14	.5512	.040	.020	204HJB	1.130	1.530	1.649	.30	204J
25	.9843	47	1.8504	12	.4724	.025	.015	105HJB	1.247	1.587	1.673	.20	105J
25	.9843	52	2.0472	15	.5906	.040	.020	205HJB	1.320	1.720	1.840	.35	205J
30	1.1811	55	2.1654	13	.5118	.040	.020	106HJB	1.511	1.869	1.978	.30	106J
30	1.1811	62	2.4409	16	.6299	.040	.020	206HJB	1.580	2.060	2.203	.55	206J
35	1.3780	62	2.4409	14	.5512	.040	.020	107HJB	1.710	2.110	2.229	.40	107J
35	1.3780	72	2.8346	17	.6693	.040	.020	207HJB	1.857	2.382	2.565	.75	207J

APPLICATIONS. Aircraft accessories, turbines, compressors, pumps and other high speed components operating at high temperatures.

DESCRIPTION. Metric series non-separable angular contact high temperature bearings for smooth, quiet operation with minimum vibration at high speed under moderate to heavy loads; support radial loads and thrust in one direction only; generally need light thrust loading.

Angular contact construction permits larger number of balls and greater load capacity than deep groove bearings of same size.

Extremely thin section but sturdy one-piece machined bronze ball retainer has high conductivity to dissipate heat. Outer ring piloted "halo" design gives increased lubricant penetration, circulation and cooling, resulting in longer bearing life at high temperatures.

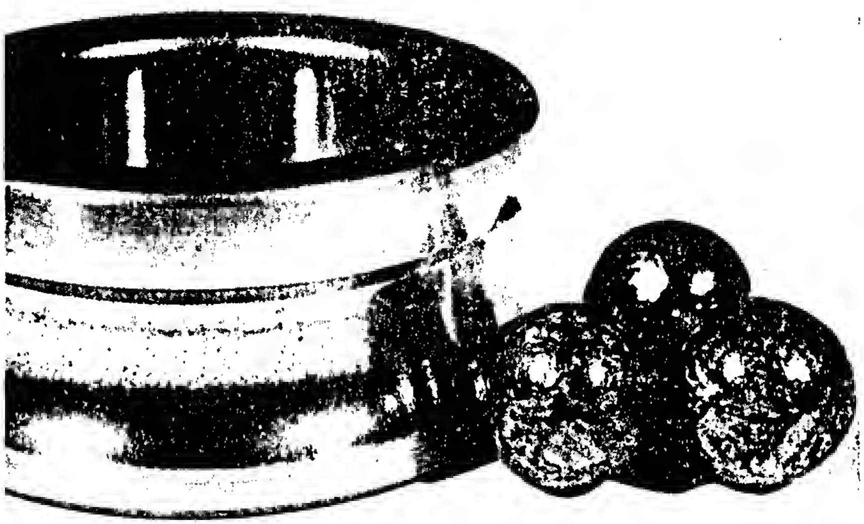
MATERIAL. Rings and balls: SAE 52100 chrome bearing steel; some sizes al available in corrosion resistant AISI 440C stainless steel. High temperature mat rials, such as M-50 tool steel, also available on special order. **Retainers:** SAE 6 bronze. Further data on page 48.

LUBRICANT NORMALLY SUPPLIED. Oil: MIL-L-6085A (Barden code 0-11); requir subsequent lubrication with continuous oil mist or spray application. Further da on page 66.

TANDEM PAIRS. All sizes listed may be obtained in DT duplex pairs for addition load capacity. Further data on page 54.

FURTHER DATA. Tolerances, page 46. Radial and axial play and yield, page 4 Shaft and housing shoulders and fits, page 68.

Exhibit 3 - 204HJB1519 Bearing After 630 Hours Operation



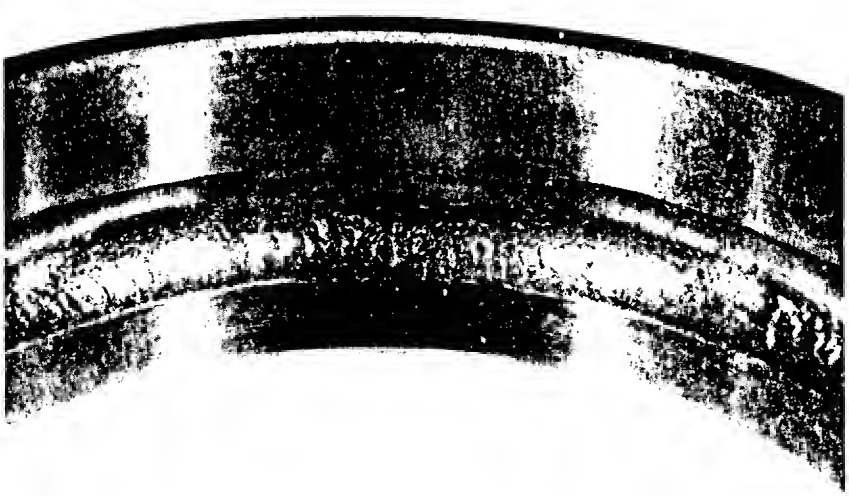


Exhibit 3 - Specification Excerpts on MIL 5606 Hydraulic Fluid

REQUIREMENTS

3.1 Qualification. The fluid furnished under this specification shall be a product which has been tested and passed the qualification inspection specified herein, and has been listed on or approved for listing on the applicable qualified products list. Any change in the formulation of an approved product shall require requalification.

3.2 Materials. The fluid shall be clear and transparent consisting of petroleum products with additive materials to improve the viscosity-temperature characteristics, resistance to oxidation, and antiwear properties of the finished product.

3.3 Petroleum base stock requirements. The properties of the petroleum base stock used in compounding the finished fluid, before the addition of any other ingredients required herein, shall be as designated in Table I when tested as specified in 4.7.2.

TABLE I. *Properties of Petroleum Base Stock*

Property	Value
Pour Point (max) [1]	—59.4° C. (—75.0° F.)
Flash Point (min)	93.3° C. (200.0° F.)
Acid or Base No. (max)	0.10
Color, ASTM Std (max)	No. 1

[1] Pour point depressant materials shall not be used.

3.3.1 *Specific gravity.* The specific gravity of the base stock shall be determined as specified in 4.7.2 but shall not be limited. Samples of base stock submitted for acceptance tests shall not vary by more than ± 0.008 at 15.6/15.6°C (60.0°F) from the specific gravity of the original sample submitted for qualification tests.

3.4 Additive materials.

3.4.1 *Viscosity-temperature coefficient improvers.* Polymeric materials may be added to the base petroleum oil in quantities not to exceed 20 percent by weight of active ingredient in order to adjust the viscosity of the finished fluid to the values specified in 3.5.

3.4.2 *Oxidation inhibitors.* Oxidation inhibitors shall be added to the base oil in quantities not to exceed 2 percent by weight.

3.4.3 *Antiwear agent.* The hydraulic fluid shall contain 0.5 ± 0.1 percent by weight of tricresyl phosphate, conforming to Specification TT-T-656.

3.5 Finished fluid. The properties of the finished fluid shall be as specified in Table II and 3.5.1 through 3.5.11.

TABLE II. *Properties of Finished Fluid*

Property	Value
Viscosity in centistokes at 54.4° C. (130° F.) (min).	10.0
Viscosity in centistokes at —40° C. (—40° F.) (max).	500
Viscosity in centistokes at —54° C. (—65° F.) (max).	3000
Pour point (max) [1]	—59.4° C. (—75.0° F.)
Flash point (min)	93.3° C. (200.0° F.)
Acid or base No. (max)	0.20

[1] Pour point depressant materials shall not be used.

3.5.1 *Color.* The fluid shall contain red dye in concentration not greater than 1 part of dye per 10,000 parts of oil by weight. There shall be no readily discernible difference in the color of the finished fluid and the standard color when tested as set forth in 4.7.3.

3.5.2 *Corrosiveness and oxidation stability.*

3.5.2.1 *Corrosiveness.* When tested as specified in 4.7.2, the change in weight of steel, aluminum alloy, magnesium alloy, and cadmium-plated steel subjected to the action of the hydraulic fluid shall be not greater than ± 0.2 milligrams per square centimeter of surface. The change in weight of copper under the same conditions shall be no greater than ± 0.6 milligram per square centimeter of surface. There shall be no pitting, etching, nor visible corrosion on the surface of the metals when viewed under magnification of 20 diameters. Any corrosion produced on the surface of the copper shall be not greater than No. 3 of the ASTM copper corrosion standards. A slight discoloration of the cadmium shall also be permitted.

3.5.2.2 *Resistance to oxidation.* When tested as specified in 4.7.2, the fluid shall not have changed more than —5 or +20 percent from the original viscosity in centistokes at 54.4°C (130.0°F) after the oxidation-corrosion test. The acid or base number shall not have increased by more than 0.20 over the acid or base number of the original sample. There shall be no evidence of separation of insoluble materials nor gumming of the fluid.

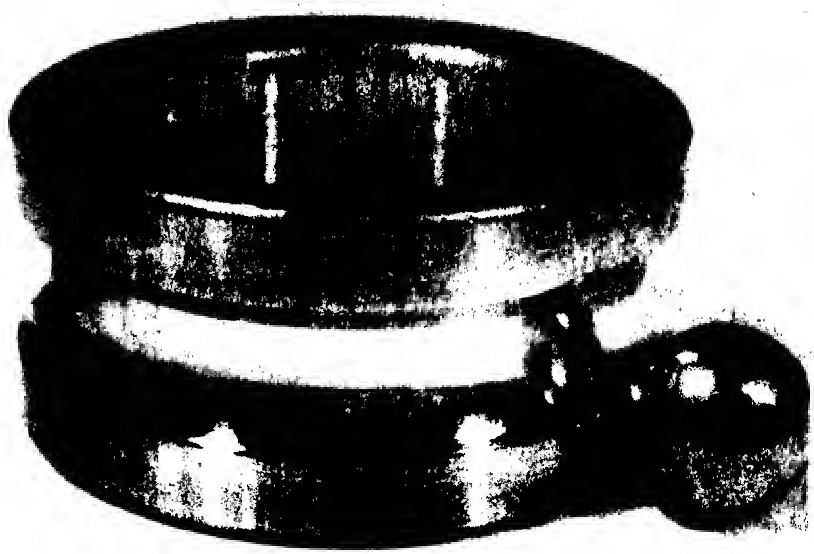
3.5.3 *Low temperature stability.* When tested as specified in 4.7.2 for 72 hours at a temperature of —54 ± 1°C (—65 ± 2°F), the fluid shall show no evidence of gelling, crystallization, solidification, or separation of ingredients. Any turbidity shall be not greater than that shown by the turbidity standard.

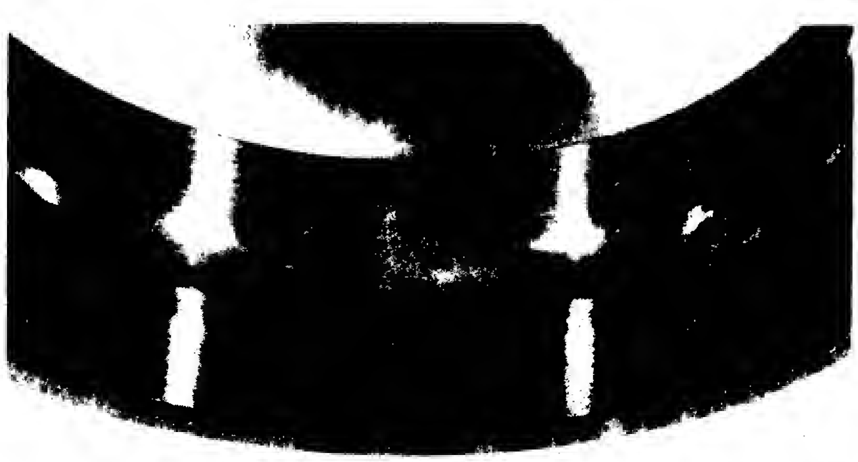
3.5.4 *Shear stability.* When tested as specified in 4.7.4 the percent viscosity decrease of the hydraulic fluid, measured in centistokes at 54.4°C (130.0°F) and at —40°C (—40°F), shall be no greater than the percentage viscosity decrease of the shear stability reference fluid nor shall the acid or base number have increased by more than 0.20 over the original acid or base number.

3.5.5 *Swelling of synthetic rubber.* When tested as specified in 4.7.2, the volume increase of the standard synthetic rubber L by the fluid shall be within the range of 19.0 to 28.0 percent.

3.5.6 *Evaporation.* The residue after evaporation for 4 hours at 65.6 ± 3°C (150 ± 5°F) shall be oily and neither hard nor tacky when tested as specified in 4.7.2.

Exhibit 4 - 204HJB1519 Bearing After 700 Hours Operation





C O P Y

Exhibit 5

Examination of Failed Bearings:

(1) First Set: 1800 hrs. Pump End Brg. Barden 204SSTX5

Inner Race: completely failed all around circle, from over center at bottom to shoulder and shoulder rolled. Pitting rather shallow, no deep spots, and over all area. No signs of heating.

Race also showed one shallow band of contact near opposite shoulder, and narrow: may be rubbing of failed parts.

Too far gone to tell initial failure point. No difficulty in bore or clamped shoulders.

Outer Ring: Outer same but smeared over pitting, probably foreign matter pits from inner as surface not nearly so rough as inner.

Balls also badly pitted, shallow type over nearly half of surface, with a few deeper pits or breaks from riding edge of inner after inner failed.

Measured one ball where not failed: .3125" indicating no wear.

(Shield rubbed on booster pump, after inner failed)

Cage: 2 piece riveted bakelite compound with alum. side plates. Broken to disassemble brg. Heavy rubbing on bore of outer, one side only and severe pocket stress with many imbedded small steel flakes. Secondary failure: inner went first.

Conclusion: Heavy load, mostly thrust with possible contribution from lubricant of poor lubricity: (indicated by peculiar type of shallow pitting.)

(2) Closed End Bearing: Barden 203SSTX5;

No failure: only surfaces badly spread with fine hard foreign matter pits. May have come from front bearing failure, but difficult to see how.

Inner Ring: Inner edge of contact fuzzy and measurements questionable. May have been two separate contact overlapped.

Bore had turned on shaft, rubbing mostly on both sides and not middle as tho shaft were hollowed. Also shoulder rub and wear.

Outer Ring: also pitted. O.D. rubbed and turned with most of contact at one end (outer end).

Balls and pressed steel cage very good.

Thomas Barish
December 9, 1963

Exhibit 5 (continued)

(3) Second Set: 700 hours Pump End: Barden 204HJB1519

Inner Ring: shallow pitting load failure as above but caught much earlier. Extended only about 180° with tapering off at each end, and only for width of contact at max.

Outer Ring shows only small pitting from hard particles off inner. Otherwise in very good condition. No difficulty on bore, O.D. or shoulders of either race.

Cage (halo bronze type) extremely good in view of inner failure. No deterioration at all. Likewise balls good. No wear and only very faintly banded.

Closed End; Second Set: Barden 203SSTX5 but shields removed.

Entire bearing in very good condition. No wear or loss of initial polish except slight greying of contact areas.

Inner showed fairly heavy load, per calculation on page 4. However contact may have been two contacts overlapped but still high and indicates more than 3 to 7 lbs. expected.

Complete absence of any pitting at all indicates that back bearing does not need shielding normally.

Thomas Barish
December 9, 1963

TASK CORPORATION II (C)

Further recommendations on the bearing problem were submitted in writing after galling appeared on the second bearing (204HJB1519) by both the Barden Company and the bearing expert, Mr. Barish. Both recommended that bearings of still higher capacity be used. A number of possible causes of failure were suggested, but there was some conflict between the opinions of Mr. Barish and those of Barden as to the reasons for failure. Mr. Wireman wondered which of the suggested causes of failure he should consider most likely and what should be done to cure it. He was especially puzzled about why the change from the first bearing, No. 204SST5 to one of higher capacity, No. 204HJB1519 had not cured the failure, since both the bearing company and Mr. Barish advocated higher capacity as the answer.

Opinions of the Barden Bearing Company

After receiving the second set of failing bearings from the first motor, the Barden Company returned its analysis in a letter dated December 4, 1963. Barden repeated its earlier injunction, that a bearing of heavier capacity was needed. The failure, Barden explained, was due to the poor lubricity (their opinion) of Skydrol. As a cure, Barden recommended that custom bearings, costing from $60 to $175, in small quantities, be used. A copy of the Barden letter appears as Exhibit 1.

Opinions of the Geyser Pump Company

Geyser Pump Company engineers said they had in the past experienced somewhat similar wear problems in bearings of pumps like that being used with the Task motor. Their answer had been to use a custom bearing of higher angular contact which had been designed by another bearing company. At the suggestion of Geyser Pump, Task contacted the company which had made the special bearing. The bearing company suggested for Task a custom bearing of higher contact angle similar to the custom bearing proposed by Barden and costing around $1000.

Opinions of Thomas Barish

The first two bearings which had failed, together with prints of the motor and pump as shown in Task Corporation (A) were sent to Mr. Barish.

Prepared in the Design Division, Department of Mechanical Engineering, Stanford University, by Karl H. Vesper as a basis for student engineering problems. Assistance from Jack Wireman, Elmer F. Ward, Dino Morelli and Thomas Barish is gratefully acknowledged.

His analysis of these bearings, which was dated December 14, 1963 and appears as Exhibit 2, was more extensive than that of Barden. He, too, suggested a number of possible causes of failure. Rather than lubricity, Mr. Barish saw the main cause of failure as being fatigue, but he also suggested several other possible sources of difficulty. Among the suggestions of his report were modifications to the design of the pump as well as the use of heavier bearings.

Mr. Barish recommended that special bearings be avoided and standard bearings used. With standard bearings, he said, there was less likelihood of unforeseen "bugs".

Meanwhile, more motors were being made and tested. The second qualification test motor was shut down and disassembled after 630 hours when metal particles began showing up in the discharge fluid. Already the front bearing, a Barden No. 204 HJB 1519 was failing seriously. Pictures of this bearing after failure appear in Exhibit 3. With failure of this bearing at 630 hours, Mr. Wireman thought the evidence more strongly than ever suggested that heavier load capacity was not the answer, but he did not know how to explain the phenomenon or what to do about it.

COPY

EXHIBIT I

THE BARDEN CORPORATION
Precision Bearings
Danbury, Connecticut

December 4, 1963

Mr. Jack Wireman
Task Corporation
1009 East Vermont
Anaheim, California

Reference: My letter of September 11, 1963 to Mr. Elmer Ward

Dear Jack:

The bearings from the hydraulic pump which you sent for analysis after 700 hours of test running have been examined. The 240 size bearing is still in serviceable condition in spite of some dirt denting including a few rather large depressions caused by hard contamination in the .--2/.005 size range. The general quantity of dirt denting, however, is less than that in the first bearing we reported on last September.

The 204HJB1519 bearing shows evidence of advanced fatigue. There are two types of fatigue patterns, the more prominent, which is fairly deep spalling, extends for approximately one half of the circumference of the inner ring contact area. There are also, in the remaining contact area, several small spots of surface fatigue. In addition, there is appreciable dirt denting which most probably resulted from the material plucked out of the raceway. There is no evidence of excessive radial unbalance or thrust loading or of improper mounting.

In comparing this bearing which failed at 700 hours with the original 204SST5 bearing which failed at 1800 hours, we find that this bearing has light traces of surface fatigue and moderately heavy spalling in the raceway while the earlier bearing showed extensive metal erosion from surface fatigue and resultant heavy wear in the bearing. In other words, in this bearing the distress that has occurred happened quite swiftly while that in the earlier bearing was the result of a much slower process over a very long time. It is possible, indeed probable, that the surface fatigue in the 204SST5 bearing had begun even before the 700 hour point was reached. We cannot explain why the surface fatigue in the 204HJB1519 bearing generated the heavy spalling while that in the other bearing continued as a surface fatigue with general wear resulting. It is however, quite apparent that the heavy fatigue does not have the same pattern as a typical, heavy load, subsurface fatigue. The fatigued ring was checked for hardness with readings of 60.75 to 61 Re with a specification tolerance of 58.5 to 62 Rc.

COPY

Mr. Jack Wireman
Task Corporation

December 4, 1963

The corrective action to take to overcome the failure condition is not changed -- to increase the capacity of the bearing so that the unit stress will be decreased to a point where the lubricating qualities of the hydraulic fluid will be adequate to sustain life for the required 3000 hours. In accordance with our recent telephone conversations we have designed a bearing within the envelope dimensions of the 204 bearing which we feel should be adequate for the application. This is the 204HJBX31 bearing. In this bearing the ball complement has been increased to ten 11/32 balls and the radial play has been increased to give a nominal contact angle of 35°. Inasmuch as the retainers in both of the above bearings were in practically new condition, there has been no change made in the retainer. The one being used is, again, 80-10-10 bronze. The increased ball complement and high radial play result in a design life of 500,000 hours at 75 lbs. thrust load. Bearings can be modified to this design in four weeks at a price of $175.00 each for six bearings or $60.00 each for 25 bearings. Prices for production quantities will be forwarded as soon as an inquiry can be processed.

The bearings are being returned herewith. Should you have any question on the above or if we can be of further assistance please don't hesitate to contact us.

Yours very truly,

THE BARDEN CORPORATION

Herbert D. Williams
Senior Product Engineer

HDW:etk

CC: Carl Berg, Purchasing Agent

COPY

For Task Corporation, 1009 East Vermont Ave., Anaheim, California
Attention: Jack Wireman, Project Engineer

Pump-Motor 56383: Failure of 204 Ball Bearing.

(1) General Conclusions: Bearings failed because of excessive thrust load: plus minor contribution from fluid not being the best lubricant;

(2) Detail Bearing examination, pages 1 and 2. First pump end bearing too far gone to draw conclusions, except primarily inner race fatigue failure: cage behaved very well. No heating from rub or bad lubrication.

Second pump bearing showed clear fatigue failure, primarily thrust load, on inner. May have been appreciable radial load.

Both small end bearings showed much larger thrust load than expected 3 to 7 lb. from spring. Areas showed 92 lb. thrust (page 1) but his is misleading as it resulted from 2 or more load paths overlapping which looked like one large load. Nevertheless the actual load was 5 to 10 times the expected: indicating outer race did not slide in aluminum housing in operation (or spring was bottomed in error).

(3) Expected Loads: page 5;

Pump spring,	40 to 70 (on test, 47)
Booster Pump	36
Spline Friction	58 to 64
Total Thrust	134 to 190 lb.

If the spline imbeds, the thrust can be very much higher. With the present design, imbedding with a sharp edge is almost certain because

(a) the unit pressure is considerable, 2190 psi projected area,
(b) the spring takes out all initial shake, and holds spline in one place with all motion at pump, (none in motor).
(c) female spline at pump too soft.

In addition, there may be considerable axial expansions due to
(a) alum. housing and steel shaft at lower or higher (same temp.),
(b) shaft temporarily much hotter than housing, and
(c) any lift-off at valve plate in pump, when operating.

Radial loads can come from misalignment, since long heavily loaded splines will have very little rocking effect. If assume spline tight, and max. runout of .004", will give 31 lb. on motor bearing.

Also magnetic loads may be appreciable. However failures indicate primarily thrust loads.

(4) Measured Loads from contact areas on the bearings, page 6 show:

169 to 197 lb. thrust on second 204 bearing

but these figures are not exact: estimate -0 to + 50%.

Thomas Barish
December 14, 1963

(5) Estimated Life: Using latest ABEC formulas, second bearing (Barden 204 HJB) has a rating of 473 lb. radial at 6000 rpm. (B10 life of 500 hours.) At 183 lb. thrust load, this gives 4200 hours B10 life.

Apparently, it took appreciably more thrust load, + some radial, + inferior lubricity to bring this down to 700 hrs.

(6) Even a loose bearing would not give appreciable radial motion to permit large magnetic loads. With thrust down to 80 lbs., max. radial motion is under .0002" at 120 lb. radial. Curves, page 7.

Recommendations:

(1) Check unit for possible axial bind. The second bearing, 204HJB showed a fairly constant heavy load, whereas spline thrust would vary.

Beck check: thru any available aperature (or make one), push shaft or rotor against spring. Should move .010 to .020, after reaching 40 to 70 lbs., without much load increase.

(2) Eliminate possibility of spline imbedding: For changeover that does not require any major parts change (except maybe truing motor shaft bore) see design 2, page 8. Need only replace quill by 2 piece design, and new longer spring. For new motors, recommend design 3.

(3) Increase thrust capacity by changing to 40° or 35° bearing. These are in regular production by MRC (7204P); SKF (7204B, maybe): New Departure (30204): and Fafnir (7204W). Be sure to obtain either bakelite-compound or bronze one-piece cages: one with lots of space to allow oil (and foreign matter) to pass thru easily.

It is not necessary to have these made by others as specials, at greatly increased cost, and with some experimentation involved. The first two makes above have 10 to 15% more ball capacity, (34 to 54% more life.)

(4) On new designs, cut diameter at booster pump next to bearing, and leave some debris space around bearing. See design 3, page 8.

(5) Improve shaft fit for smaller (203) bearing. One bearing returned showed turning and loose fit. Needs tighter fit when no locknut. Use mfr standards, and be sure bearing slightly loose to allow for press fit. Recommend mfrrs "loose fit" standard.

(6) Believe steel liner in housing will be necessary for small bearing. In spite of many efforts, no one seems to have been able to make this work without a sleeve for long life and any but very small loads.

In the meantime, for current units, recommend .0003" loose min.

(7) Review airgap selection; Petroff's equation shows about 3 h.p. loss in gap even at 2 centifpoises viscosity and .007" gap. At larger gap, less fluid loss may offset magnetic losses.

Thomas Barish
December 14, 1963

COPY

Load Calculations:

Thrust Load: (1) Preload Spring on Quill: 40 to 70 lbs.
Actual on Test Units 47 lb.

(2) Friction from Spline:

11.5 H.P. x 63000/6000 rpm = 120.7 in. lb.

Tangential force, 120.7 / .23" radius = 524 lb.

Axial Friction at .11 to .16 coef. of friction

= 58 to 84 lb.

This number may be exceeded if spline tends to imbed.

Spline Loading: Area: .46" p.d. x .60" long = .28 sq. in.
(tangential projected area)

Unit Loading: 524 / .28 = 2190 psi:

("if equally divided.)

(3) Thrust from Booster Pump:

2.8" O.D. at 6000 rpm = 73.3 ft./sec.

Hydraulic head at 80% eff. V^2 x .8/ 2g = 67 ft. head

= 16 psi

Thrust on seal diam., 1.7" = 36 lb.

Total Thrust: 134 lb. to 190 lb. with possible large increase if spline imbeds.

Radial Load: 1/2 rotor weight + 1 G unbalance = 10 lbs.

Possible load from eccentricity if splines imbed:

(or from pump housing tilt)

For .004" eccentricity (.002" radius)
on beam 2 1/2 long x .34" diam. (ends fixed) 31 lb.

Thomas Barish
December 9, 1963

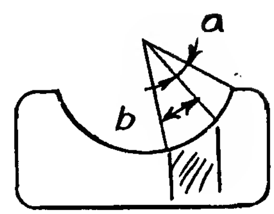
COPY

Load Calculation from Brg Contact Measurements:

	First Set - 1800 Hrs.		Second Set - 700 Hrs.	
Barden Brg. No.	204SS-TX5	203SS	204HJB1519	203SS
Balls	8 -5/16	8-17/64	11 - 5/16	8-17/64

Data from Barden

Curvatures %	52-52	52-52=	52-54	52-52
Inner Groove Rad.	.1625	.1382	.1625	
Groove Diam.	1.0175	.8564	1.0175	
Land Diam.	1.1300	.9520	1.1311	
Groove Depth	.0562	.0478	..0568	
degrees	48.5	48.5	49.5	
Radial Looseness	.0005-9	.0005-9	.0015-19	
Initial Contact Angle	11.47-15.85	12.5-16.8	16.25-18.3	
Hardness-Inner			60.75-61 C.	



Measurements and Calculations (inner races)

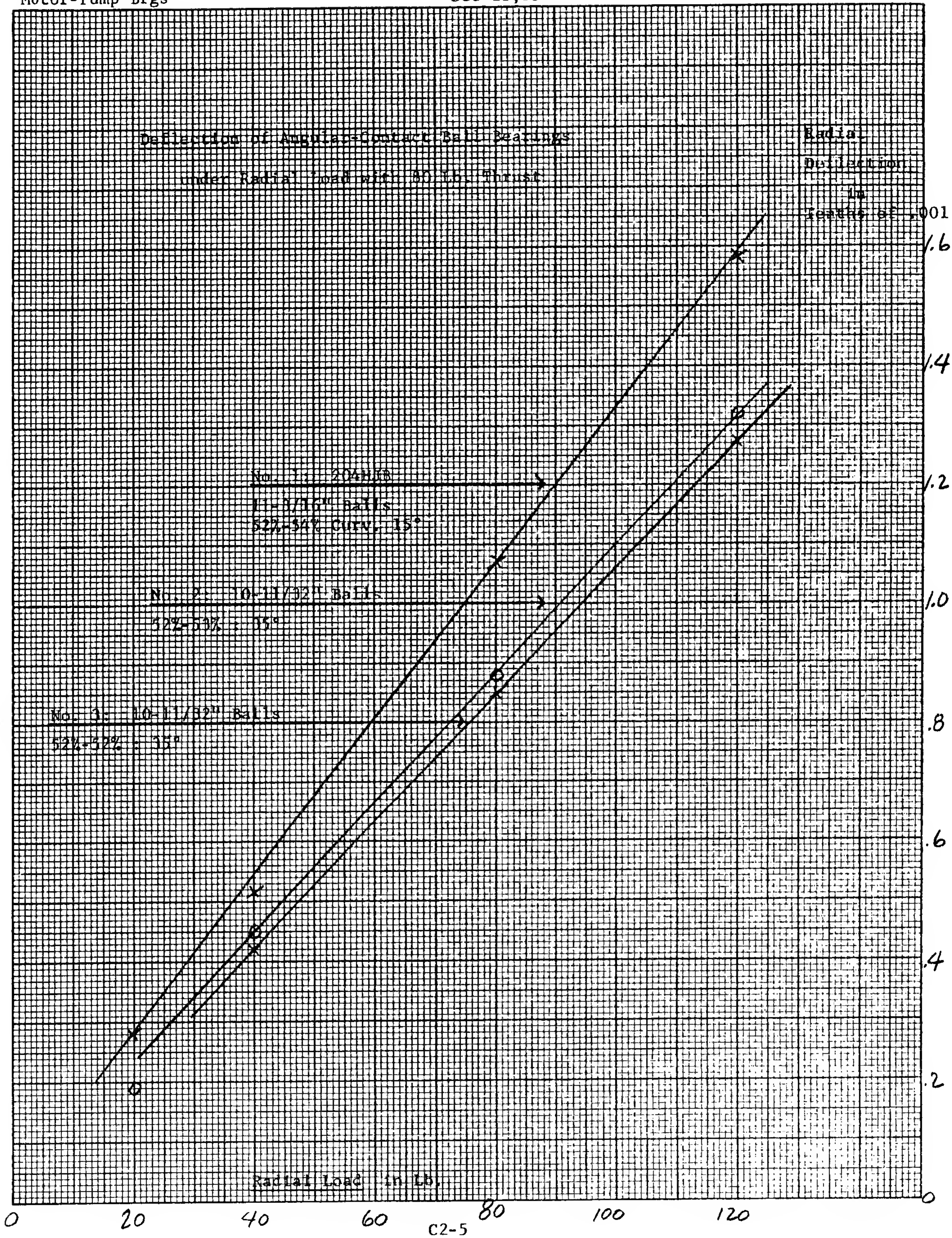
Contact a	.050	.050	.06	.050
b	.060	.060	.050	.060
Load/Ball (from b)	46	43	55	46
a + 1/2 b	.08	..08	.085	.08
in degrees	34.	28.6	30.	34.
Contact Angle	14.5	20.9	19.0	14.5
Thrust Load	92 lb.*	169 lb.	197 lb.	92 lb.*

Contact angle, calculated using measured load and initial angle

T/nd^2K	.00153	.00148	.000860	.00153
angle	18-20.4	18.5-21.2	19.6-21.3	18.5-21.2

(* Both of these may have been two or more paths overlapped)

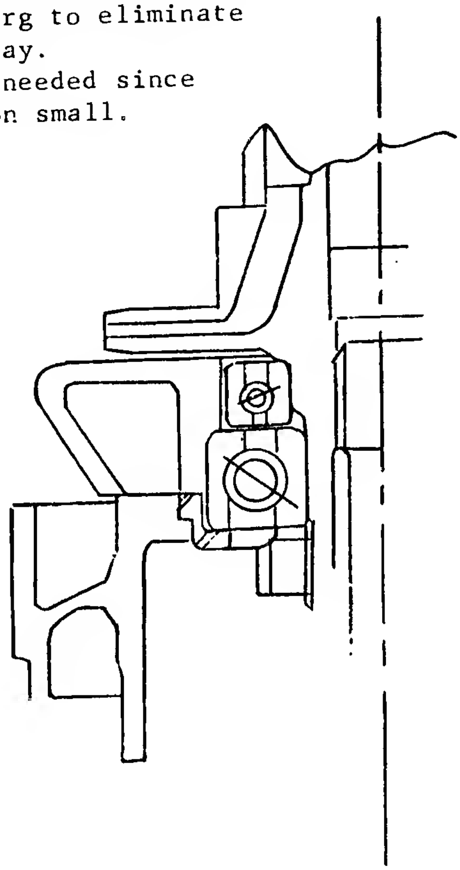
Thomas Barish
December 9, 1963


Deflection of Angular-Contact Ball Bearings
under Radial Load with 80 Lb. Thrust
Radial
Deflection
in
Tenths of .001
No. 1:
11-3/16" Balls
52%-54% Curv. 15°
No. 2: 10-11/32" Balls
52%-53% ; 35°
No. 3: 10-11/32" Balls
52%-52% ; 35°
Radial Load in Lb.
0
20
40
60
80
100
120
.2
.4
.6
.8
1.0
1.2
1.4
1.6

Sketches for Correction.

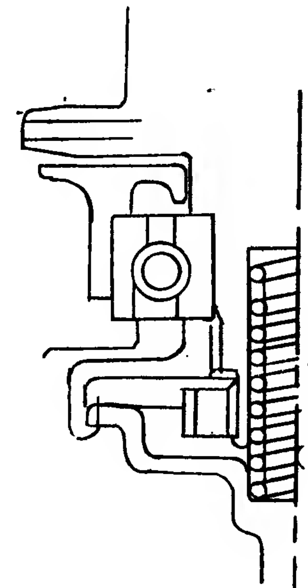
Design 1:

Method for using 1905R as preload Brg to eliminate radial play.
Not needed since deflection small.



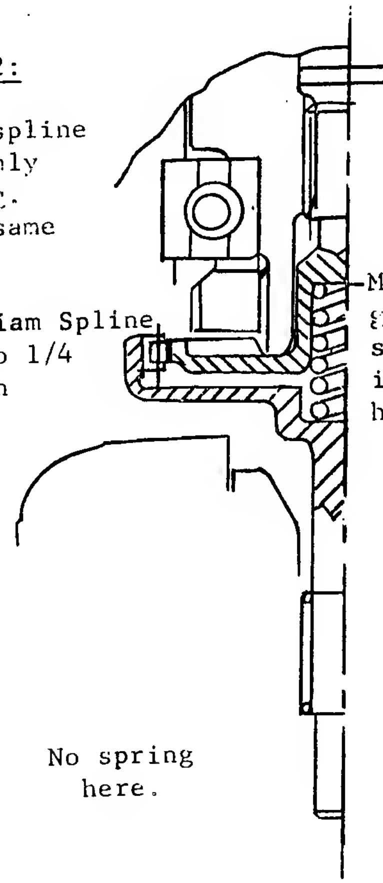
Design No. 3:

For new motors.



Design 2:

To eliminate spline imbedding Only change in cplg.
Rest of unit same



TASK CORPORATION II (D)

Following failure of the third bearing at 630 hours, several further actions were taken by Task to determine the cause and cure. Additional bearings were sent to Mr. Barish, who examined them and supplemented his earlier reports with more possible explanations and reiteration of some of his earlier recommendations. A second consultant, Professor Dino Morelli of the California Institute of Technology, was asked to examine the bearings and give his opinions. Professor Morelli said he thought the cause was skidding of the balls due to too light an axial loading for the particular type of bearing running fully immersed in fluid. The recommendations of Professor Morelli differed from those of Mr. Barish, leaving Mr. Wireman with the question of which of the recommendations should be followed, if any.

Further Opinions of Mr. Barish

The third set of bearings was sent to Mr. Barish for examination. In addition, at his request, additional sets were copper-plated and run for ten hours, one set on a pump and one set on the Task Dynamometer with no pump, after which they were sent to Mr. Barish. From impressions on the copper, it could be seen where the balls were riding in the races and how wide was the band of contact, which in turn enabled him to compute axial shaft loadings.

A report on the third failure and an analysis of the copper-plated bearings was submitted by Mr. Barish on December 26. Some possible causes of failure suggested in this report were, (1) binding between housing and rear bearing and housing, and (2) axial vibrations due to the hydraulic pump. From the copper-plated bearings, Mr. Barish concluded that axial forces on the shaft were within the predicted range, though the pump imposed a somewhat higher thrust than was to be expected from the loading spring alone. This report appears as Exhibit 1.

Another report dated December 31, by Mr. Barish, gives his findings for examination of five additional bearing sets which had run in service from 10 to 20 hours. In this analysis, which appears as Exhibit 2, Mr. Barish cited binding or misalignment of the rear bearing as the most likely source of difficulty.

Prepared in the Design Division, Department of Mechanical Engineering, Stanford University, by Karl H. Vesper as a basis for student engineering problems. Assistance from Jack Wireman, Elmer F. Ward, Dino Morelli and Thomas Barish is gratefully acknowledged.

Still later in correspondence, Mr. Barish made the following comments:

"I apparently lost the needed emphasis on the major problems in trying to cover all possibilities and also because the first reports were made without seeing parts or bearings.

"Major Problems: friction in the spline produces thrust load, estimated at 54 to 68 pounds. This always occurs, whether there is imbedding or not. Imbedding can cause much greater loading.

"The spline problem is aggravated because the spring held the 'floating' coupling tight one-way, and even small movements would start to build up thrust. Also the small diameter and greater length mean higher loads and little alignment capacity.

"Second: Usual practice uses steel liner in aluminum housings for ball bearing outer rings.

"Temperature changes and temperature differentials cause radial binding between outer ring and housing on the small bearing. Hence, there is much more thrust until the bearing is free to slide. Even transients here can be destructive."

A characteristic of the bearings which was discussed with Mr.Barish over the telephone was that of "ball banding," which had appeared on some of the service units' bearings. On some of the bearings, each ball had acquired a narrow circumferential stripe of wear, indicating that it had rolled in a continuously repetitive pattern. When some of the bearings had been taken for copper plating to the Bearing Inspection Company, a specialized Los Angeles testing laboratory, Mr. Wireman had discussed the ball banding with representatives of the laboratory. They told him the banding showed that overloading of the bearing had prevented the balls from moving in a random fashion. Mr. Wireman asked Mr. Barish about this on the phone and was told that ball banding always occurred under operation with continuous thrust and was not an indication of overload.

Bearing Inspection also informed Mr. Wireman that bearing problems on pumps such as that being used with the Task motor were not uncommon. In other pumps having the same general design and operational characteristics there had been problems of failure of bearings within the pumps. Mr. Wireman understood that the approach taken to solve such problems had been to install bearings of higher load capacity.

Opinions of Professor Morelli

Ball skidding due to a combination of factors was seen by Professor Morelli as the main cause of failure. These factors were as follows:

1. By operating fully immersed, a retarding force of viscous drag was imposed on the balls as they passed around the race. Catalogs on bearings, it was pointed out, specify "mist lubrications" for ball bearings. (In contrast to Professor Morelli, Mr. Barish saw full immersion of the bearings as an advantage. In his opinion, it assured full lubrication and the continuous flow tended to reduce the danger of localized heating.)

2. Being "outer race centered" there was an additional retarding force on the balls exerted by the retainer as it dragged through viscous shear against the stationary outer race. The point at which the retainer contacted each ball also caused it to apply a force inward on the balls lowering the contact pressure between the balls and the outer race.

Professor Morelli concluded that the answer was to increase ball contact stress, to use an "inner race centered" bearing, or to modify the design so the bearings would not be running fully immersed. His recommendations were to use an inner race centered bearing of load capacity comparable to that of the bearing originally used.

C O P Y

Exhibit 1

for the Task Corporation, 1009 E. Vermont Ave., Anaheim Calif.

Attn. Jack Wireman, Project Engineer

Supplement A to Report of Dec. 14, 1963 on

Motor Bearing Failures in Motor-Pump 56383.

(1) Examination of the new failure, SN/5, Barden 204HJB (page A3) is helpful because failure had not progressed so far. It showed that failure started with the balls riding on the edge of the groove. The ball surface breaks were typical.

The races did not have any real surface breaks, only severe roughening: except outer did begin some slight surface breaks.

Severe roughening extended over entire grooves.

(2) The balls could ride the edge of the inner under two conditions: the first a very severe tilt of the outer, of the order of .005 to .010"/". The outer race surface roughening was over all the race except a few long islands in the center (sketch, page A3). This also agrees with bad tilt. Likewise cage deterioration with severe ball-pocket wear and light wear between cage and outer.

(3) The second possibility (and I believe more likely) is that the outer race of the small brg was bound in the aluminum housing and did not slide: happens if aluminum housing is cooler than shaft and brg. Then also, the outer housing would shrink endwise and bind small brg against large end brg. Practically every small end brg showed much larger thrusts than spring preload. (109 lb. this set, page A3)

On the first failure (conrad brg) this reverse load on the 204S would be less harmful; than on later 204 hjb. Here, the reversing would quickly cause ball to ride on shallow shoulder (about .008" end movement) and this would break up the ball surface rapidly.

The new failure rode on both edges of outer path. Brg would then break loose and bounce back to other side: showing clear islands again. Also cage could show the same distress, mostly from broken ball surfaces. Also inner contact extended about 20° over-center towards wrong side.

(4) Other possible causes:
Critical frequencies, axial: first using pump spring, = 1700 rpm. Too low to matter. Second using brg spring rate, pages A5 and A6 give critical of 38,000 per min. Still low, since one-per-cylinder forcing function is 54000.

Shock loads, when small brg lets go quickly, only rise to 660 lb.; small compared to shock capacity (brinell load) of over 2000.

Note discussion on plated brgs, page

C O P Y

Pump-Motor 56383

Recommendations:

(1) Same as before: check carefully for errors in handling, chips, binding etc. in assembling pump and motor.

(2) As before: eliminate spline imbedding by two-piece coupling per sketches. This will also isolate pump further from motor for eccentricities, transfer of forcing vibrations, and will permit longer preload Spring.

(3) Increase thrust capacity on 204 as before.

(4) Add much emphasis on eliminating bind in 203S outer race fit. A good steel liner, and looser fits are indicated. In view of trouble, recommend about .0004" min.

A good changeover is available with present parts by changing from 203 to 103 size leaving room for .10" thick liner.

Plated Brg experiments proved very helpful: The contacts were all where they should be and showed no trouble (measurements and calculations, page A4) except that load on small brg was large as on all previous cases. Also the no. 3 brg (small end) shows a slight misalignment of outer race, and smaller load.

The positive conclusion that can be drawn from this is: these brgs would not have failed if continued as they were. The failures came from conditions introduced by some other operating conditions such as cold starts or unusual temperature differentials: or else by some assembly or machining errors existing on other units.

Thomas Barish Dec. 26, 1963

Examination of Additional Failed Brgs: S/N No. 5 Motor- 600 Hrs.

(1) Brg. Barden 204HJB1519. This brg shed new light on the problem. Both races were fairly completely covered by very bad roughing up of the surface with only some parts of outer groove breaking thru the surface slightly. This exisited over all the outer, and on inner from shoulder to about 20° over center. Both races had the ball run hard enough on the shoulder edge to raise a positive burr.

In addition the outer race showed some long islands, about at the middle of the roughened area, and about 1/5 of the width where the surface was relatively smooth. (sketch below)

Most of the balls were badly broken on the surface and some of them along a diam, about 180°, and much deeper than the race surfaces. This means that the balls rode the edge of the groove and the breaks were started by this edge cutting, (both on inner and outer).

This also means that the initial failure was on the ball surfaces from riding this edge and that the race roughening resulted from the ball failure.

In addition, the outer showed signs of having rotated slightly, on both O.D. and thrust shoulder.

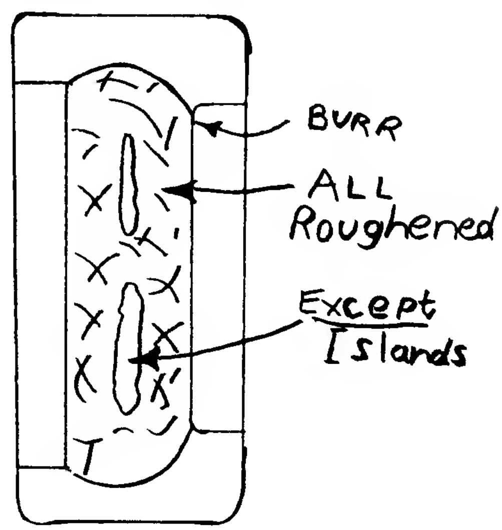
The above condition, wide path with clear area in center of outer, is produced by (1)* a badly cocked outer race plus thrust load. Reference- "Effect of Misalignment on Forces acting on Cage" by K. Kakuta, ASME Paper 63-Lub-12-.

This was confirmed by the unusual retainer deterioration. All the ball-pocket surface showed bad spreading, some very severe with high burrs: whereas the O.D.-Outer-ring contact showed only a little distress. It was thought that unequal ball size might account for the variation in pocket wear but enough of the balls could be measured on unbroken diameters to show no wear (all .3125) and all equal.

Also the cage enlarged slightly under severe forces and bound slightly on outer ring.

Normally. such a tilted outer would show a bad tilt at the ball path, instead of the wide spread. But in this case, the outer turned somewhat in the housing.

There were no signs on inner or outer of appreciable chips under the thrust shoulders, or uneven clamping.



(2) The small Brg: Barden 203S showed no failure. Contact area measurements, a = .045": b = .06" (page 6, 1st report) which indicates 109 lb. thrust or two overlapping paths. Thrust was right way.

Bore showed no contact except one corner, and also appreciable rub at shoulder. Again, too loose a fit on shaft and probably badly tapered. Outer showed no motion in housing.

*(2) Alternate: rear brg outer sticking in housing, building up reverse thrust and then letting go. Perhaps building up thrust in opposite direction.

Thomas Barish Dec. 26, 1963

C O P Y

Examination of Copper-Plated Test Brgs

These brgs all showed just what might be expected (except for larger thrust on small end brg) and nothing like what had shown on failed 204 size. In each case a fairly narrow uniform width of contact. Measurements and calculations follow:(fig. page 6, 1st report)

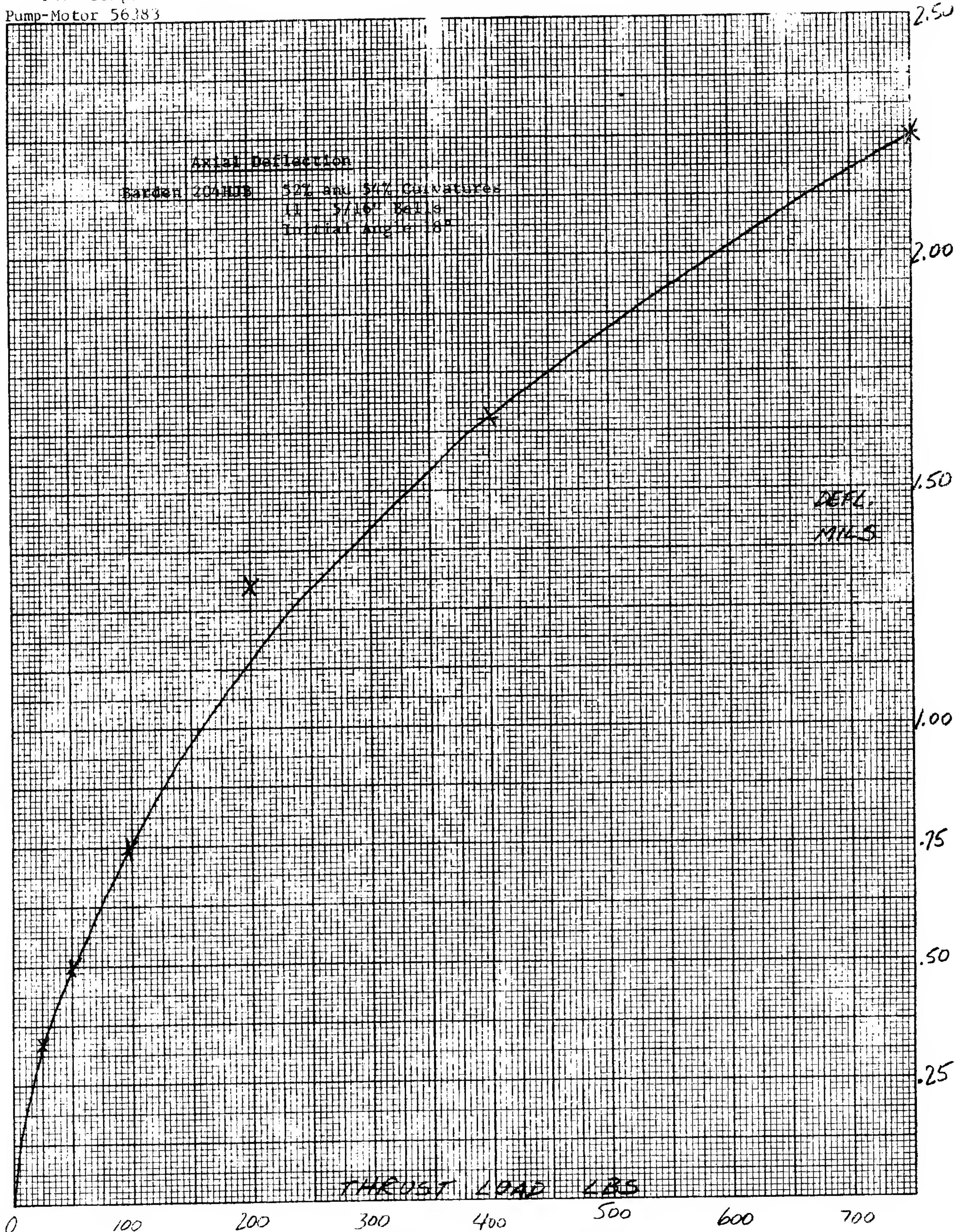
	With Spring alone		With pump	
Brg. No	no. 2	no. 3	no.1	no. 4
Size	204HJB	203S	204HJB	203S
inner:a,b		.045 .058		.050 .050
at 90^{o}		.045 .055		" "
		.045 .058		" "
		.045 .060		" "
average	.0620.046	.045-.0578	.060-.058	.050-.050
Outer:a,b		.030-.050		.065-.035
		.038-.050		.065-.035
		.050-.042		.057-.042
		.045-.050		.060-.038
average	.068-.035	.0408-.048	.060-.042	.0618-.0375
Load/ball				
Inner	19	40	38	30
Outer	20	26	33	12
1/2a+b: inner	30^{o}	30.6	31.3	31.2
outer	29.2	27	27.5	35.4
Contact angle	19.5^{o}	17.9	18.2	17.4
Inner	20.2	21.5	22	13.4
Outer				
Measured (initial by BII)	18.3	15.0	18.0	15.5
Total Thrust				
Inner	69.5	98	131	72
Outer	76	76	128	22 **

Conclusions: Thrust on main brg about where they should be: (note about 73 lb. spring alone, and 130 with pump)

Thrust on small brg always larger than expected.

** (These readings show small but positive outer tilt.)

Thomas Barish Dec.26,1963


Axial Deflection
Barden 204HDB
52% and 54% Curvatures
(11 - 5/16" Balls
Initial Angle 18°
2.50
2.00
1.50
DEFL.
MILS
1.00
.75
.50
.25
THRUST LOAD LBS
0
100
200
300
400
500
600
700

COPY

Other Possible Causes:

(1) Critical Frequencies: axial: Rotor Weight, 5.5lb.

Spring Constant taken from curve page A5, for 100 lb. load

$\simeq$ 187,000 Lb. per inch.

Deflection under own weight = 5.5 / 187,000 = .000 0294

Critical Frequency = 197.7 / $(.000\ 0294)^{1/2}$ = 38000 per min.**

Forcing functions: 6000 for rpm and 54,000 for 9 cylinders.

(2) Shock Effect: if rear brg hangs up and lets go quickly:

Total Travel: estimated at .012"max. (endplay of 204 brg.)

Energy build up: 70 lb. spring x .012"

Using area under deflection curve, page A5, this would require brg force to go from 100 lb. to 660 lb. max. shock

This is still not excessive since brinell capacity or shock capacity of the brg is over 2000 lb.

** (Note: 35° Brg will increase spring rate about 60 to 70% and will raise this critical to 48,000 to 50,000 per min.)

Thomas Barish Dec.26,1963

C O P Y

for the Task Corporation, 1009 East Vermont, Anaheim, California
Attention: Jack Wireman, Proj. Engr.

Motor-Pump 56383: Bearing Failures.

Supplement B to Report of Dec. 14, 1963: Review of 5 additional Brgs. Barden 204HJB1519 that had seen small amounts of use.

(1) Bearing examination on page B2. No deterioration. But ball paths just distinguishable in 4 cases.

Conclusions:

(2) The thrust loads were about where they should be, perhaps a little high, but nothing to cause any trouble.

(3) There were no signs of any tilt or misalignment of ball path.

(4) However two of the bearings (and slightly in a third) showed that at some time, the small end bearing took most of the thrust and left this large end bearing under pure radial load; in one case leaning slightly towards reverse thrust.

(5) This confirms previous analysis (emphasized in Supplement A) that the major cause of trouble is seizing of outer race of the small end bearing preventing axial sliding.

And that this trouble is aggravated in longer runs by small heavily loaded spline held in one place and tending to imbed:

And by possible eccentricities at Pump-Motor joint causing larger radial load.

(6) One possible alternate for this lot; if motor was run without pump or spring load, bearings would show radial load as above.

(7) Previous discussion only considered as source for binding of O.D. of small bearing, the possible greater temperature of shaft and bearing over housing. One other possibility now suggested is that unsymmetrical end-bell under fairly large pressure load distorts bearing housing out-of-round.

(8) Recommendations still as in Supplement A:

Thomas Barish
December 31, 1963

Examination of Bearings:

5 Barden 204 HJB1519 that had seen small amounts of miscellaneous use. (numbered 1 to 5 by TB)

None of the bearings showed failure: practically no deterioration. In fact, they could be reassembled after very light polishing, with new (high quality) balls and used as new.

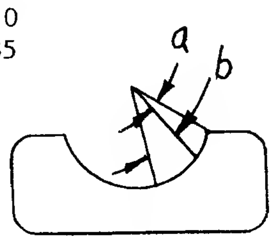
The cages were all in practically perfect condition. The balls all showed very light banding; one set, except 2 sets in bearing 2. The bands were very faint in bearings 4 and 5.

The races showed no signs of any motion on bores, O.D. or sides. However, the ball paths could be distinquished and were measured and checked for load below: (except no. 4 where marks were too faint.)

The major point of interest was that bearings 2 and 3 showed beside the standard thrust load path, a second lighter area, a small scraping on the inners, under approximately radial load. In no. 3, this even edged slightly towards the wrong side thrust. No. 5 also showed a very faint spot of radial load on the inner. On the outer races, these were not discernable because the polish was not so high and it would take more running to show these faint paths.

Ball path measurements and
load calculations

Bearing No.	1	2	3	5
Inner Race a	.060	.060	.060	.060
b	.050	.050	.045	.040
a + 1/2b	.085	.085	.0825	.080
Contact Angle, degrees	19.5	19.5	20.5	21.3
Load per Ball Lb.	22 1/2	22 1/2	17	22 1/2
Total Thrust Lb.	83	83	66	90
Outer Race a	.055	.065	.065	can't see
b	.040	.035	.040	
a + 1/2b	.075	.0825	.085	
Contact Angle, degrees	23	20.5	19.5	
Load per ball	28	20	28	
Total Thrust, Lb.	120	78	102	



Thomas Barish
December 31, 1963

TASK CORPORATION II (E)

To test the opinions advanced by the different experts as to why the bearings had failed, Mr. Wireman conducted several investigations. In his judgment, the results of these investigations cast doubt on all the opinions advanced and verified none. Some action, however, had to be taken to solve the problem. Mr. Wireman rejected the idea of trying several different bearings at once. Such an approach seemed to him "excessively expensive and sloppy engineering." His decision was to try another standard bearing, New Departure No. 030203, which had a higher load capacity, a contact angle of 35°, larger balls, and an inner-race centered plastic retainer. The thickness of the retainer was great enough so its point of contact on the balls was halfway between the inner and outer races. The 030204 was manufactured to only a standard A.B.E.C. class 3 quality, a class compared to A.B.E.C. class 7 for the Barden bearings, and its price consequently was around $3. as compared to around $10. for the Barden bearings.

As of February 24, 1964, a test motor had run with the 030204 bearing for 600 hours, and no failure had yet occurred. The main question in Mr. Wireman's mind was what he would do if the bearing did fail before the required 2,500 hour test was completed.

The possibility that the rear bearing was misaligned or binding was eliminated by careful measurement of the parts concerned while operating and computations as to the expansion effects to be expected with higher temperatures. Computations indicated that the aluminum motor housing could be expected to expand from 0.003 to 0.006 inches farther axially than the shaft as temperature rose to that of normal operation. Mr. Wireman concluded that there was more than enough room for this much axial movement of the rear bearing, and movement would be made easier at higher temperature since the aluminum housing would expand radially more than the outer race.

Careful examination revealed no evidence ofspline imbedding. Axial loading of the shaft under operating conditions was checked by measuring how far the shaft moved with the motor running. Exhibit 1 shows schematically how this measurement was made. No excessive deflection resulted, and it was therefore concluded that the maximum load on the shaft was not excessive.

Varying the applied axial loading by varying throttling of the hydraulic fluid outflow, the shaft was forced to move about 0.008 inches back and forth with the motor running and it was observed that the rear

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Prepared in the Design Division, Department of Mechanical Engineering, Stanford University, by Karl H. Vesper as a basis for student engineering problems. Assistance from Jack Wireman, Elmer F. Ward, Dino Morelli and Thomas Barish is gratefully acknowledged.

bearing slid longitudinally in the housing, as it was supposed to, without binding. Observation through a plexiglas cover which was substituted over the rear bearing made it possible to see that the belleville load was sufficient to keep the rear bearing balls from skidding.

The possibility of ball skidding of the front bearing was also studied. The front of the motor was replaced with a transparent plastic through which the bearing could be seen. The motor was then filled with oil, and run. With stroboscopic light, it was possible to tell by observing the retainer whether the balls were revolving at the right speed or going too slow and skidding. By varying the axial load hydraulically, the loading at which skidding occurred could be measured. It was found that the 204HJB1519 bearing would not skid unless the axial load dropped below 60 or 70 pounds. With the lighter load 204SST5 bearings, only 20 to 30 pounds were needed to prevent skid. With the 030204, around 40 pounds were needed. All these figures were less than the loading expected in operation, so Mr. Wireman concluded that ball skidding was not the cause of failure.

Mr. Wireman also looked for signs of cavitation in the hydraulic fluid, but he could not see any. He surmised that the pressure of 50 psi in the motor was enough to prevent cavitation.

These tests were performed on the Task Dynamometer. It was not possible to conduct them with a pump attached to the motor, but this fact, in Mr. Wireman's opinion, did not cast doubt on the validity of the results.

Another possibility which had occurred to Mr. Wireman was that there might be some sort of high frequency pressure feedback from the Geyser pumps. He had inquired with Geyser about this and been told that no such feedback was present. He commented, however: "We've been told that bearing problems have occurred with that type of pump before. The pump wails like a banshee when it's running. Something has to be vibrating quite a bit."

If the bearings were failing due to excessive loading imposed by the pump, it was presumed that Geyser Pump might be liable for the expenses of determining the cause of the bearing problem and for correcting it. Such liability would have to be determined on a negotiated basis, however, since the original specification to which the motor had been designed had made no reference to axial loadings on the motor shaft due to the pump.

As of mid-March, 1964, all the pump motors were being assembled with the New Departure 030204 bearing. A test motor was also being run in the Task shop with this bearing, and after 600 hours no problems had yet occurred. It had been decided that all the bearings should be "sound tested"[1] by the Bearing Inspection Company before installation. At first

1. In "sound testing" an unloaded ball bearing is spun next to a microphone. The noise level produced is compared to a standard. Excessive noise indicates roughness. Such testing costs about $1 per bearing, depending on quantity.

a rejection rate of 50% was experienced with this inspection, The cause was found to be that the supply house from which the bearings were bought had been repackaging them with insufficient care. In repackaging, some of the bearings had apparently become exposed to moisture which caused slight corrosion and consequent roughness of the balls and races.

Aircraft were flying with the pumps installed, some having the Barden 204HJB1519 and some having the New Departure unit. Maintenance instructions on the aircraft were set to require tear down and inspection of the pump motors after 1,000 hours operation, but none had yet been in use this long. A couple pumps had failed during pre-flight testing due to clogged outlet filters (10 micron filters), but none had yet suffered bearing failures. Some other problems with the motors, such as burning of the electrical connector pins, need for higher flowrate, faster cold starting and lower current drain had been corrected by various design modifications, none of which were expected to affect shaft bearing life. Not all of these changes were Task's responsibility, and the costs of making them had accordingly been divided among Task, Geyser Pump, and Thunder Aircraft by mutual agreements.

Over 120 motors had now been shipped, and it had been agreed among Task, Geyser Pump, and Thunder that Task was responsible for performance of the bearings. There were 180 motors yet to be completed and shipped on the contract.

Exhibit 1 - Schematic of Axial Movement Checker